AIR ENGINES

Volume One

CONTENTS

INTERVIEW WITH PAUL TRENTHAM, CONSULTING ENGINEER
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4/14/04

I called Paul Trentham, a consulting engineer in Joplin, Missouri because I had his name in connection with the Terry Miller/Toby Butterfield air car project from the previous decade. I wanted to find out if he knew the fate of the project, and especially what happened to the three air cars I knew of that Terry had built. The interview was not recorded, but I have summarized it below based on notes from our conversation.

Paul Trentham is a consulting engineer, among whose accomplishments is that he worked on the Apollo engines for the moon shot in the 1960s.

Paul got involved with the Joplin air car project about the time Terry got out of it. (Editor's note: Terry was the inventor, and Toby his business partner; Terry gave his share to his daughter, Teresa Robbins. Terry and Toby are now deceased.) Toby called Paul and explained that he and Terry had come to an impasse, that the partnership was not working out, and that Terry had lost interest in pursuing it. Paul was asked to make an analysis of the invention to determine if there was any practical use in continuing it. Paul only met Terry once, and was not well acquainted with Ms. Robbins, Terry's daughter. His dealings were with Toby.

Regarding the air cars that Terry built, Paul recalls only the blue Chevy Sprint (editor's note: there was also the famed "Air Car One," a three-wheeled workbench on wheels that Terry showed from coast to coast (1980s), and a red compact car, probably either a Chevy Sprint or a Ford Festiva (1990s, called "Miss. Rexroth," the blue car was the "Spirit of Joplin")). Paul met Teresa Robbins, Terry's daughter, at a meeting where the project collapsed and everyone went their separate ways. Ms. Robbins has moved away from the Joplin area. The project had no tangible assets at the time of its dissolution, and the blue car was mashed for scrap. Paul never saw Air Car One. He still has a number of high pressure air tanks, originally made for CNG usage, and would like to sell them. Most have never been used.

Paul wrote his report in about 1992 or 1993. The good news was that he found a definite use for compressed air as a motive power for vehicles. The bad news was that he felt Terry's way of going about it was all wrong. In Paul's opinion,

which was based on computer models and other research, multiple expansion is impractical because it can only be optimized at one set of conditions: one speed, one cutoff. He is convinced that the 2nd, 3rd, and 4th expansion stages on Terry's engine were actually dragging on the 1st stage, instead of augmenting its power. Paul tested the car with a simple Prony Brake placed on a front driving wheel, the other driving wheel on the ground. The power of the engine was measured and found to be pulsating and unsteady, falling to zero and rising to about 1.5 hp. This seemed reasonable with the knowledge that all of Terry's cars needed pushing to get them started, even on a flat surface. He felt that compound engines could work well for a steady speed stationary engine, but less so for locomotion, and certainly not more than two stages total. He cited Norfolk and Western Railroad efforts with two stage compound steam locomotives based on the Mallet design.originating in France. The compound idea was well known in The Steam Age, but never more than two stages in locomotives, and for good reason.

He felt it would be much more practical, in a vehicle engine, to use a single-stage engine with variable cut off. Some air is used inefficiently at long cut offs required for acceleration from zero, but the time is so short the loss is not worth the extra complexity of multiple compounding. The most economical use of air seems to be at a very short cut off of only 5%. To realize high efficiency at such a short cutoff requires near zero clearance space in the engine cylinders, and very good seals. Rotary valves help to reduce unwanted clearance space, and have no reciprocating mass problems causing vibration and wasted power loss.

Toby asked Paul to design a better engine, which he did. He determined that the most efficient cutoff point would be 5%, but in order to start and accelerate the car, something like 50-60% would be needed. At this cutoff range, the engine would develop enormous torque, as in steam engines, and no transmission would be needed. (Note that Terry's vehicles also had no transmission, but never achieved much torque or power due to the ineffective and complicated multiple compounding scheme.)

Paul designed and built a variable cutoff engine over the next three years, converting a Honda 500cc V-twin motorcycle engine to run on air. He moved the equipment from the Joplin air station to his home and built a garage and shop to house the project. He invented a rotary valve to eliminate complicated reciprocating valve parts, which allowed faster engine speeds than the old-fashioned expansion engines were capable of, and a simple cut off control valve. One rotary valve took care of intake and exhaust, (flow to and from the cylinder) while another small variable cut off valve ahead of the intake/exhaust valve adjusted the cutoff (the amount of stoke during which the inlet valve was open.) Although he was not happy with the seals on the valves, it did work. It developed about 9.5 horsepower at 5% cutoff, and was operated up to 800 psi. Higher pressures were not used due to the leaky valve seals.

He accidentally discovered that the engine would function well at up to 14,700 rpm, by opening the engine full throttle from rest, with no load. The engine jumped almost immediately to that speed, and ran at that speed with almost no vibration. The compressed air was entering the engine at the pressure tank's ambient temperature of 83° F., but he had trouble measuring the exhaust temperature at 5% cutoff because the expanded air bottomed out his temperature gauge, which couldn't go below -75°. He has a VHS tape somewhere, showing that if you blew across the engine's exhaust stream, the moisture in your breath would freeze and fall to the ground as snow. The engine ran smoothly down to about 30 rpm.

The University of Washington, Seattle, got interested in the engine. The Aerospace Engineering Department had been working on a liquid nitrogen car, which used a marine auxiliary steam engine they had found and adapted for their purposes by installing it to run a small mail truck. They invented and patented a triple-pass heat exchanger that warmed liquid nitrogen up to nearly ambient temperature, using the free energy of the atmosphere to do the warming. Paul spoke very highly of their heat exchanger, saying that it did not freeze up and functioned very well. After trying the former steam engine, they had some grant money left, so Paul sold the engine he had developed to the U of W and they put it in their cryogenic car. The young professor working on the project was Dr. Carl Knowlen, whose address is U of W College of Engineering, Aerospace Building Room 136, Seattle, WA. The head of the department is Dr. Adam Bruckner, and his predecessor, Dr. Abe Hertzberg was the founder of the nitrogen car project. Dr. Knowlen wrote SAE papers on the car, but the project ran out of grants.

Paul questioned the use of liquid nitrogen by the U of W project, rather than liquid air. The U of W was concerned about using liquid air because they thought the oxygen would separate and cause a fire hazard in an accident. Paul says that since the molecular weights of oxygen and nitrogen are very close, the cost of a stirring device to keep them from separating in storage would be insignificant. Dr. Knowlen studied the use of cryogenic air and nitrogen and the economics and safety issues, and delivered a paper at the Society of Automobile Engineers conference in Los Angeles. This was during the popularity of battery powered cars, so it did not get a lot of attention.

Paul still thinks that air is the way to go. He never got to finish what he wanted to do with his engine. He is now working on a rotary valve for internal combustion engines, with fully variable timing. He has been granted two patents for his rotary valve and has another patent in the works.

The seals on the original short cutoff engine limited the pressure and therefore the effectiveness and efficiency of that engine. A different approach on the seals would be used if further air engine designs were undertaken.

Paul notes that a cryogenic car can be run on ordinary compressed air by bypassing the heat exchanger, although the inverse is not true. He states that the cryogenic car has the advantage over the compressed air car, in that it can store 2.5 times more air in a given tank at a given pressure. Therefore the cryogenic car can exceed the range of the pneumatic car by more than twice. The cost of doing that is to have a U of W type heat exchanger on board to warm the cryogenic air up to near ambient, with the "free" ambient energy. It is this ambient energy converted back to useful driving energy that extends the driving range of the car by double over the compressed air concept. Another way of stating it is that the car can carry twice as much mass of air as a liquid, but it all comes out to the same impressive gain in on board energy storage. You get air conditioning as a side benefit whether the car is running on cryogenic air or highly pressurized air.

Paul is still interested in working on air engines and would be willing to be hired on to do engineering work on either cryogenic or pneumatic systems. He strongly believes in the superiority of the cryogenic system, which would need to be supported by tanker trucks transporting liquid air to cryogenic air stations, because of the greater range between fill-ups compared to the compressed air car.

Paul's own written explanation of the advantage of cryogenic-over-compressed air is reproduced here:

> "Liquid air is much denser than pressurized air. We were using lightweight tanks designed for and used by the CNG (compressed natural gas) cars. CNG burns cleaner than gasoline, and is used in some places, but has much lower energy availability, generates Carbon Dioxide, etc., so it probably won't ever be THE solution. These tanks are good for, I believe, 4000 psi. Because cryogenic air (liquid air) is a liquid, instead of a gas, it is much denser, or about 2.5 times as heavy as even the highly pressurized air at around 4,000 psi. (I hope my numbers are right, it has been several years since I went through this). You see, air is quite light at atmospheric pressure, but it gets more dense as you pressurize it. If you pressurize it to two atmospheres, that is to say, about 30 psi instead of 14.7 psi, then it is twice as dense, because you have twice as many molecules in the same volume. And so forth. And when you go from high pressure (about 3500 or 4,000 psi) to liquid, you get about 2.5 times as much liquid air in the same volume tank that held the high pressure air. Above 4,000 psi, it is not practical to pressurize air and haul it around in a tank, it is generally agreed it is too dangerous if ruptured, it is like a damn big gun exploding. So that is sort of the agreed on upper limit for hauling around pressure vessels of anything safely on the highways, air included. Plus the tanks get to be very heavy walled, and expensive, and weigh a lot.

“Liquid air is made in large plants by pressurizing the air, and then removing the heat caused from the pressurization, and then doing it again and again, until it collapses from the gaseous state to a liquid, and it is very cold, I can't remember, lousy memory, but it is around 200 degrees F below zero. Now it is not under great pressure as a liquid, but will vaporize and turn back to gaseous air as ambient heat leaks in through the tank, and that causes the pressure to rise, so you have to insulate the container, and that is often done by making a Dewar flask, which is like a thermos bottle. Fortunately there are some really good insulators from the past few years, and you can insulate a tank with stuff like glass foams that do a great job of keeping the liquid air cold. Even so, there is a little heat leaks in, and I think they figure on losing about 5% in a week, due to heat leaks, and having to vent the excess pressure off.

"Dr. Knowlen and I looked at this several years ago and he wrote a paper on it, so you guys might call Washington U in Seattle and ask him for a copy. I got one somewhere, but it is filed away in the junk, but I remember that we found out from the big suppliers that it costs about 15 cents a gallon to make liquid air. Carl figured it out for reasonable mileage and storage, and I think it came out that liquid air was the equivalent, economically, of operating a gasoline powered car at $1.30 per gallon. So you see, the economics are not at all bad, and no gas price wars with the Arabs and Californians driven by environmentalists to not build more refineries, etc.

"Liquid cryogens, mostly Oxygen, Nitrogen, Argon, and CO_2, are hauled all over the country in those long shiny tank tractor trailers, and I have never heard of a bad accident like you have with gasoline tankers. The big liquid air plants are expensive to build, so they haul their product hundreds of miles. They crack off Oxygen, Nitrogen, Argon, etc. from these plants much like you do in a petroleum refinery, based on their liquification temperature. They don't explode, not even Oxygen, because there is no fuel to ignite when the tank breaks. Remember, the pressure is not high!!! Oh the oxygen will burn anything organic it touches, like gasoline, that is how the Moon rockets worked, and you don't need a spark plug. But by and large it is safer to store liquid air than it is to store gasoline, in my opinion.

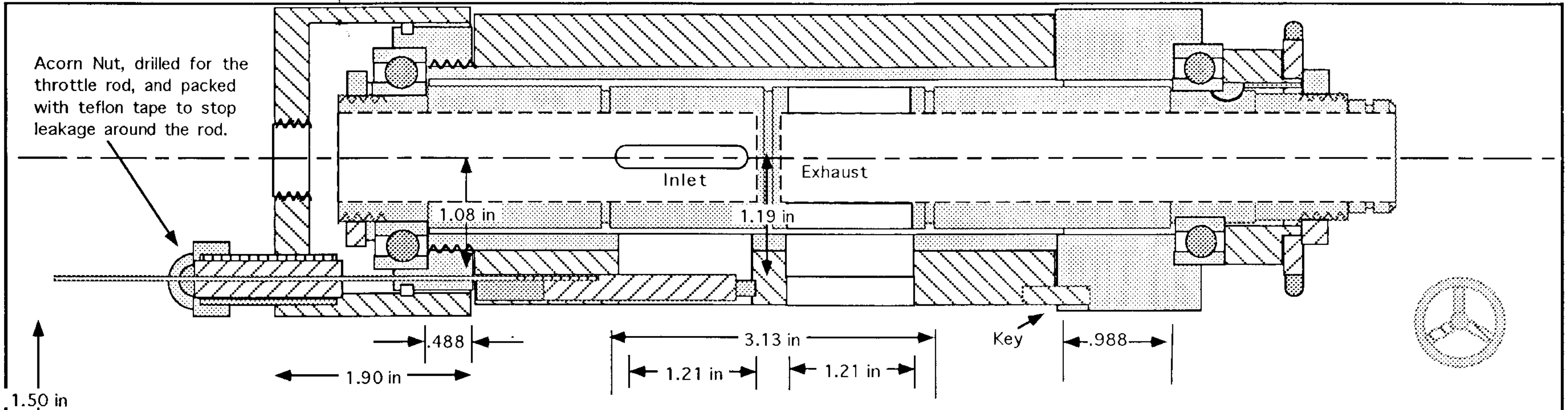
"You could have filling stations just like you have now, only they would have insulated liquid air tanks. They get filled once a week by the big highway tankers. Special nozzles used to fill the liquid air tank in your car, not a big deal, and not at high pressure like the

pressurized air tanks. If you have a leak, so what? The liquid air quickly vaporizes, and turns into air, and it is much cleaner than the air in LA, because liquifying it takes out all the dirty stuff. Neat, hunh?

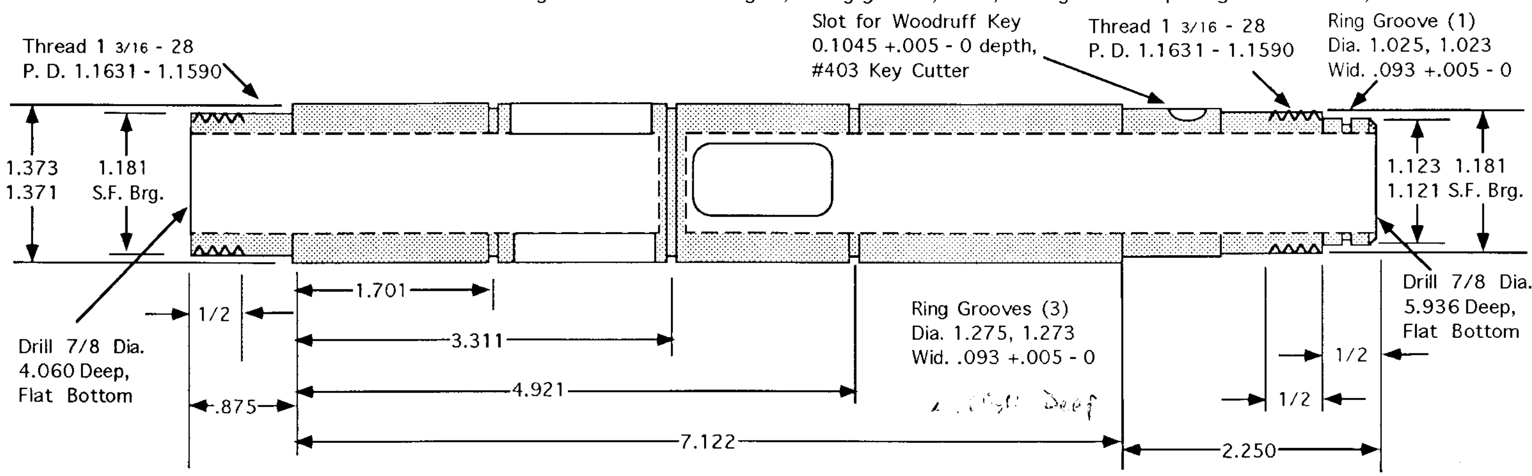
"Still won't have as good mileage range as gasoline powered cars, but about double what you will have in pressurized air cars. And if you are remotely located and don't have a liquified air station, just use your handy dandy high pressure air compressor and run the car at half its liquid air range. The liquid air tank will probably have the same pressure rating as the high pressure air tank.

"A word about air conditioning. You don't get the AC from the very cold liquid air, you get it from the expansion of the air in the cylinder. So you get A/C whether the car is liquid air or pressurized air tank. I was taking air at about 80 degrees F from a pressurized tank and putting it through my engine at several hundred psi, and it was expanding to do work in the cylinder and coming out the exhaust at below 75 degrees below zero and you could make it snow by blowing your moisture laden breath across the exhaust. My seals were not very god, so I only ran at 300 to 700 psi, and that is not very efficient use of the air, but the engine ran as high as 14,700 rpm, and, by the way, there was almost NO vibration, and it would run nicely down to about 30 rpm. You do not need much if any transmission in a "steam" (Rankine thermodynamic cycle) engine. Lots of torque when you need it for short time, though inefficient at long cut offs.

"Cryogenic air doubles your tank range, and it is safer than high pressure air. You can use the same vehicle and tank for high pressure air, it is just a little bigger with the insulation. I would rather hit a cryogenic tank full of liquid air, than a 4,000 psi air tank. The difference in explosive power and potential hazard in an accident is enormous. Spilled cryogenic air on asphalt, might or might not burn the asphalt, I don't think so, as 80 percent of the liquid air is Nitrogen, and non flammable. It boils off to become air very rapidly. That is why you want to use liquid air, rather than liquid oxygen, and you don't need to go all the way to liquid Nitrogen to make it safe, but some tests would be needed."



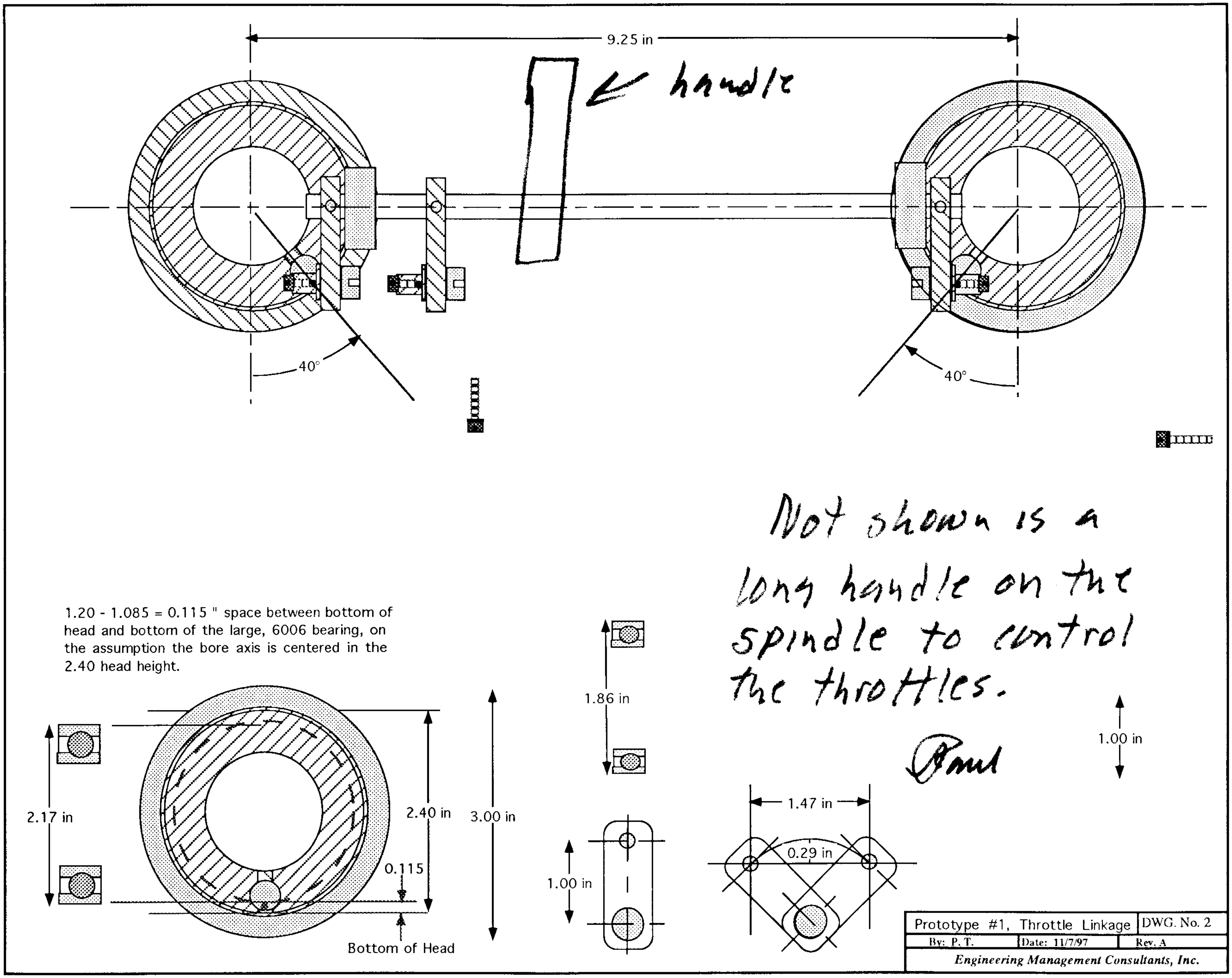
This is the first "In Head" throttle design actually built for testing in January, 1998. The blue crosshatched sleeve is not drawn exactly, but does show schematically how the throttle operates. The blue crosshatched sleeve fits inside a smooth bore in the actual head, and is static with the head, being positioned precisely over the milled inlet and exhaust slots in the head. That arrangement allows us to use the same head with various trials on the blue crosshatched sleeve so we don't have to re-do the head every time we try something new. Also, we can take the sleeve and valve assembly out of the head and work on it without taking the head off the engine, saving gaskets, labor, sealing the fluid passages in the head, etc.



Prototype #1, Machined Parts	DWG. No. 2	
By: P. T.	Date: 11/7/97	Rev. A
Engineering Management Consultants, Inc.		

Carl, we got rid of the cap, and reduced the undesireable volume in the inlet side with a smaller bearing in an improved design, but have not built it yet. This works, but is awkward.

Paul



9.25 in
handle
40°
40°
Not shown is a long handle on the spindle to control the throttles.
Paul
1.20 - 1.085 = 0.115 " space between bottom of head and bottom of the large, 6006 bearing, on the assumption the bore axis is centered in the 2.40 head height.
2.17 in
2.40 in
3.00 in
0.115
Bottom of Head
1.86 in
1.00 in
1.47 in
0.29 in
1.00 in
Prototype #1, Throttle Linkage
DWG. No. 2
By: P. T.
Date: 11/7/97
Rev. A
Engineering Management Consultants, Inc.

From knowlen@aa.washington.edu Wed Feb 23 10:25:03 2000
Date: Wed, 27 Oct 1999 14:07:05 -0700 (PDT)
From: Carl Knowlen <knowlen@aa.washington.edu>
To: LN2 -- abe hertzberg <plueagle@aol.com>,
Adam Bruckner <bruckner@aa.washington.edu>,
Carl Knowlen <knowlen@aa.washington.edu>,
Tom Mattick <mattick@aa.washington.edu>
Subject: LN2 leaks 2 of 2

---------- Forwarded message ----------
Date: Thu, 26 Aug 1999 16:48:20 -0500
From: Paul Trentham <tigertrentham@4state.com>
To: Dave Scanlan <daves@afmusa.com>, Carl Knowlen <knowlen@aa.washington.edu>
Subject: Re: Seals for UW Rotary valve application

Dave,

I was pleased at your quick response. I have tried to reach Dr. Carl Knowlen, UW in Seattle, who is our contact on that project, but he is out. We freqently correspond by e-mail or phone. UW has completed two DOE research grants on this project in the last 4 years, but currently have no research funding to continue work on it. Their physical effort has been to design and install a "vaporizer", which converts the cryogenic fluid to a gas. This is accomplished by using the warmth of ambient air to "boil" or vaporize the cold cryogen fluid, and it accomplishes this without freezing up the heat exchanger with the moisture in the warm ambient air. They hav converted a small postal van as a "test mule", and have our rotary valve engine, or "expander", installed in it. They have an insulated flask on board to contain the cryogenic fluid, LN2, and also their "vaporizer", to provide the gas phase at pressure to our engine. It works, and they drive it around the UW campus in Seattle.

We sold them the engine last year before we had completed development of the seals. It works quite well, but the seals we used were not adequate, and should be replaced and re-designed. This approach is unique, in that it uses the warmth of the ambient air to recover energy removed from the working fluid in the cryogenic cooling process. As a an added benefit in some applications, the gas gets quite cold again as it expands through the engine, and provides on board air conditioning as a natural consequence without further energy usage except for a fan and heat exchanger.

Dr. Knowlen and others in the Aerospace and Energetics department at UW have written several papers on this concept. His most recent paper was presented last week at the SAE meeting in Costa Mesa. It is available through SAE or UW. In this paper, He explores the economics and physics of using Liquid Air rather than Liquid Nitrogen, as is currently being used in the "test mule", and the theoretical results are encouraging. The economics are also quite encouraging, as it turns out that the cryogenic cost is dependent primarily on the cost of electricity. The cryogenic suppliers routinely transport liquid Nitrogen and Oxygen in bulk in insulated transport trailers hundreds of miles. Storage in dewars apprears to be quite safe, and the leakage or boil off can be quite low.

We hope to continue this research with UW, and need to improve the seals as one of the objectives in the next steps to be taken. Dr. Knowlen has also investigated adding heat to the cryogen by burning small quantities of fossil fuel. This approach provides an LEV or Low Emission Vehicle, compared to the ZEV, or Zero Emission Vehicle up to now. You may know that the Battery Cars are, for all intents and purposes, dead. They just did not work out for a number of reasons. Currently, Hybrid vehicles are receiving the most attention, and the LEV using some fossil fuel, CNG, or gasoline, based on the cryogen system looks very competitive. Burning a fossil fuel to heat a gas is much more simple and efficient in extracting the heat value of the fuel than in any 4 stroke engine as the other Hybrids are doing, and the exhaust emissions

X

are much lower.

To provide you with some dimensional data, I can tell you that the UW engine rotary valve diameter is 1.373/1.371 inches, with three ring seal grooves 1.275/1.273 inches, and 0.093 +0.005 -0 inches wide. The bore this valve rotates in is polished, and 1.375. Rotary valve and bore are aluminum. We are curently using 15% Graphite 85% Teflon solid rectangular section rings as seals, and the a dry film lubricant.

We are also experiencing considerable leakage at a stainless steel piston control valve. It has no seal at present, is 0.250 inch dia., and was made for a slip fit in an aluminum bore. We need a seal, or seals for it also.

We can provide some oil lubrication on the low pressure side of the rotary valve and the slideing piston control valve, if necessary. We have used Mobil One sythetic with no problems in the present engine.

The piston valve moves very slowly and only occaisionally. The rotary valve rotates at speeds up to 7,000 RPM. Pleas note that eventually the seals experience higher temperatures if the added heat of the LEV approach is used. I expect that those seals may require something similar to current usage in Turbochargers, where the first seal exposed to hot exhaust gases in simlar to a piston ring, followed downstream (lower pressure) by a hot oil seal. We are using that concept in our latest work, and provide oil cooling at the second seal to cool the seal and the rotary valve surface it seals against.

Some very interesting industrial applications have arisen. I am not at liberty to discuss them at this time. We are very close to being able to disclose more proprietary information involving patents, but may require and non-disclosure agreement for the interim.

We look forward to working with you, if you think you can help. We expect some re-machining would be necessary in the existing seal grooves, new grooves on the control piston, and reworking the valve bore and control piston bore on the UW engine, and possibly some provision for seals lubrication.

I hope this is helpful in uderstanding our needs.

Paul Trentham, P.E.

Dave Scanlan wrote:

> Paul, we have many seals in many applications. I believe we will be able to
> help you if we can get more information on your problem. Our company is
> familiar with difficult applications and we are familiar with large volume
> of parts. I would be interested in looking at the application that you
> spoke of at the U of WA. We can handle AutoCAD DWG files or DXF format.
> Could you please give us your address or phone and we can start on your
> project. Dave Scanlan 206-575-1998 206-575-2122 fax
> daves@afmusa.com
>
> -----Original Message-----
> From: Paul Trentham <tigertrentham@4state.com>
> To: daves@afmusa.com <daves@afmusa.com>
> Date: Wednesday, August 25, 1999 12:22 PM
> Subject: Seals for Rotary vlve application
>
> >I have need for a number of circular seals for rotary valves. The
> >valves distribute air at temperatures ranging from -100 degrees C to
> >ambient. The seal fits in a groove on the O.D. of hollow stainless
> >steel cylinders approximately 2 inches diameter. (detailed DWGS can be
> >provided if you can provide the seals.) The inside pressure of the
> >cylinder is about 50 bar (800 psi) and is released into the seal cavity

> >during rotation of the cylinder. The seal purpose is to prevent leakage
> >from slim clearance annulus between the rotating valve cylinder OD and
> >the bore in which it is rotating. The clearance annulus is at ambient
> >pressure.
> >
> >We have a similar application, except the annulus is filled with hot
> >lubricating oil, and the pressure is 100 bar. High pressure gas
> >temperature is 150 C, and is petroleum combustion exhaust. Plurpose of
> >this seal is minimize leakage of the high pressure gas into the oil
> >filled annulus.
> >
> >Very high production quantities of these seals appear likely. We are
> >in a development program in support of a unique engine concept which is
> >being patented. One aspect of it is being tested at University of
> >Washington in Seattle as part of a Zero Emissions Vehicle program using
> >cryogenic air. That engine needs improved seals, which we hope you can
> >provide.
> >
> >If you have, or are interested in providing seals as outlined above,
> >please contact me.
> >
> >Paul Trentham, P. E.
> >
> >

From knowlen@aa.washington.edu Wed Feb 23 10:24:51 2000
Date: Wed, 27 Oct 1999 14:05:44 -0700 (PDT)
From: Carl Knowlen <knowlen@aa.washington.edu>
To: LN2 -- abe hertzberg <plueagle@aol.com>,
Adam Bruckner <bruckner@aa.washington.edu>,
Carl Knowlen <knowlen@aa.washington.edu>,
Tom Mattick <mattick@aa.washington.edu>
Subject: LN2 leaks 1 of 2

---------- Forwarded message ----------
Date: Tue, 10 Aug 1999 11:31:53 -0500
From: Paul Trentham <tigertrentham@4state.com>
To: Carl Knowlen <knowlen@aa.washington.edu>
Subject: Re: greetings

Dear Carl,

Sorry about your lightning zapped system, and hope everything is working without any "twitching". I have been looking at the drawings of the UW expander to see how (1) the leaks can be fixed, and (2) how a variable cut off can be added, and (3) what changes would be required to use a target pressure of say 1000 psi. Perhaps you can get a lab machinist working on the leaks, with a minimum of your time and help from me advising.

(1) Leaks: I have all the possible leakage pasages identified, and have a plan to fix them. The biggest leak is at the back of the throttle cylinder. The throttle cylinder bore exits the head, and is barely covered by the head sleeve cap and the external polished Aluminum cap. We need to add a threaded plug in the outside end of the throttle bore, with a hole through it for the throttle drive wire. With a close fit, hole to wire diameter, leakage should be very small, but you can pack the inside of the pressure end if you like to further seal it. I don't know why one side is leaking, and the other is not, irregularity in the sealing surfaces probably, so fix both sides anyway.

The rest of the leaks are theroetically solvable with a circular seal. We tried a nearly solid rectangular section Teflon and 15 % Graphite seal, but these are what is in the engine now, and they are not very effective, and should be replaced with a new type of seal. The worst case is the single seal groove in the middle outside of the valve between intake and exhaust ports. Leakage here goes directly to exhaust in a short path, and you can hear it "hiss" through the open exhaust port. I wish there was axial room to put a second and third seal at that location, but there is not.

What is required to improve this seal and the other circular seals on the valve is to abandon the Graphite Teflon, and make the seals from either Buna N, (I think brake cylinder seals are made from this), or pure Teflon. Standard hydraulic seals may be available in the size you need, or modifiable, or you can machine them from Teflon with small chevron skirts that will flex slightly under pressure to provide closing force around the sealing surface. Teflon should provide improved wear under rotating conditions, and your lab machinist can machine it directly from stock. This is the same cross section geometry of the typical rubber pressure seals in hydraulic systems, for example the brake cylinders on cars with the old drum brakes. The only difference here is that we also have the surface rotating in the seal, so it is more like a shaft oil seal. You will note that we have no oil supplied at present to the present seals, because we were trying to do it with a dry lubricant on "hardcoated" Aluminum surface, but that has not worked very well. The seals proposed herein will probably require a more fluid lubricant, such as a low temperature grease or oil. An oil drip into the sealing space should work quite well. Bore and tap a hole through the top of the head into the seal space, put in a few drops of oil, and then screw the plug back in. If it works well, it will not use much oil. The seal lip triangle should be small and rather thin, on the order of 0.005 or

0.010 thousandths of an inch. Be guided by the form and size of the typical brake cylinder seals. It should be thin enough to bend and conform to the seal surface, yet long enough radially not to be blown double back through the clearance space between the two sealing parts. Care should be taken in assembly to get them in the right direction according to pressure, and to get then in the bore without folding them over.

I know that you do not have a Student (Academically Indentured Servant!) to work on this, but I doubt that you need one. Most of them would just be in the Machinist's way for what is required, because what is really needed now is a good lab machinist and a stick of Teflon. What I have described should make good sense to your lab machinist, and he may be able to cut and try some seals as described if you give him the idea and leave him alone to work on it. (Most good machinists are like that, in my experience. Many of them consider Engineers and such as a real pain in the ass, and mostly in their experience, that is for good reason.) It is not likely that the seal will have much drag even at high pressure, because the surface area of the flexible lip is so small you cannot get much pressure force on it to lock down the surface under torque. It doesn't have to be a perfect seal, but a very good seal will do nicely.

(2.) Variable Cut Off: I know that Dr. Bruckner in particular wanted Variable cut off, and it is a good idea. There are some problems however. High Cut Off uses gas at a high rate, and that means that (a), the gas supply line to the expander heads has to be appropriately large to carry the higher mass flow without significant friction and pressure drop. It also requires (b) the ability to generate the gas at that rate in the vaporizer (boiler), or to have an auxillary high pressure storage chamber filled with max pressure. Problem with that is the auxillary storage chamber volume has to be about 10x the size of the mass of high flow gas intended to be used in order to minimize (to 10 %) the wasteful pressure drop within the auxillary chamber . Similarly, if you are going to generate the high flow rather than store it, the vaporizer (boiler) has to be oversized to the same extent as the high mass flow. If you want to double the mas flow, even for a short time, the vaporizer has to be twice as big, (all the time). The best practical alternative is to carry a propane torch to augment the vaporizer, or to use as a superheater. At any rate, for the present engine let us assume the present one half inch gas delivery lines to be unchanged, though the engine could use considerably larger lines.

The Variable Cut Off requires some attention to the variable resultant head port opening size and shape. In general, the range of variable port size can be smaller with higher pressure, and all of the variation shouldbe controlled in such a way that any downstream area and volume is reduced compared to upstream, such that pressure drops are minimized. This leaves friction losses and the fluid dynamics gets somewhat simpler, but it is primarily a slot flow system with periodic interruptions. I do not know how to analyze that, and need some guidance. I expect this is common knowledge in aerodynamics, as slot flow in the subsonic range of our engine inlets should be well understood. Basically, I am narrowing a 0.20 inch wide slot for Cut Off , and closing the 1.25 inch long length of the same slot for Throttle. If this is a bad slot shape to use for up to 1,000 psi, let me know. Also, what mass flow can be expected with reasonably small friction losses in such a slot if it is wide open. Finally, how thin can it be made before it stalls, "wiredrawing" the old steam engine designers called it. I have three different schemes to consider for cut off control when I have some sense of these flow conditions.

(3.) 1,000 psi Engine: This depends almost entirely on the seals described in (1.) The rest of the engine can take it easily, except for the large polished aluminum caps. Note that they are fastened to the head with two capscrews. I haven't stressed it out, but I think they must resist about 3 or 4 tons of force at 1,000 psi. I do not want anybody hurt, so check that out and strengthen the caps to head connection if it is necessary. I have had it up to 800 psi before I realized that, and it didn't break, but I

didn't do it again. With good seals, the cap should not pressurize, so you could drill a hole in it (say, quarter inch dia. at the bottom so it drains) to make it safe, and also to listen for leaks in that area. Also check out the front cap, but I believe it is safe as is.

In summary, fix the seals first, enlighten me regarding fluid dynamics for 1,000 psi, and we may add a variable cut off after we have the seals satisfactory.

I will not be going to Costa Mesa after all. Good luck with Allied Signal meeting. I think we have a potential winner for deep mining, and need some research money to prove out the system working on cryogenic Air. It should be cheaper than pumping all that air and Diesel exhaust they have to contend with now, and a lot cooler.

Give my regards to all, especially Dr. Hertzberg.

Paul Trentham

ENERGY SYSTEMS

TRANSPORT FOR THE FUTURE: THE COMPRESSED AIR CAR!

THE CONCEPT

Just as any normal car runs on petrol or diesel, this car runs on air! The key components (equivalent to the motor car engine) are: (1) a ROTARY PNEUMATIC MOTOR and (2) a POWER PACK that consists of a PRESSURE RECEIVER and a HEAT PIPE HEAT EXCHANGER SYSTEM.

By utilizing lightweight materials in the construction of the air storage vessels, combined with a ROTARY PNEUMATIC MOTOR which accommodates a variable expansion ratio according to the inlet/outlet pressure and a HEAT PIPE HEAT EXCHANGER SYSTEM which recovers some of the energy lost in expansion, the compressed air car is as practical as any ZERO EMISSION vehicle likely to be offered on the market in the near future.

WHAT ARE ITS (MINIMUM) SPECIFICATIONS?

- POWER PACK ENERGY DENSITY: 25 WATT HOURS/KG
- POWER PACK LIFE: 1000 CYCLES
- POWER PACK COST: £6-7000
- POWER PACK RECHARGE TIME: 4-6 HOURS
- MOTOR POWER OUTPUT: 25 KW
- MOTOR POWER TO WEIGHT RATIO: 500 W/KG

The COMPRESSED AIR CAR has a target motor output of 25 kW. This allows it to achieve TOP speeds of up to 90 km/h and a range of 80-100 km over flat terrain.

THE PRESSURE RECEIVER

The function of the PRESSURE RECEIVER is to store the compressed air. The working pressure of the receiver is 33 MPa as this is the highest

pressure for which commercial COMPRESSORS are available. For maximum strength to weight ratio, the vessel will be manufactured from an advanced fibre composite as aluminum or steel are too heavy.

It would weigh about 185 kg. When fully charged the vessel would contain 33 MJ of energy at a temperature of 25°C, with the weight of the air approximately 330 kg. Depending on the type of compressor used it is possible to charge the vessel in anything ranging from a few hours to 15 minutes!

PNEUMATIC MOTOR DESIGN

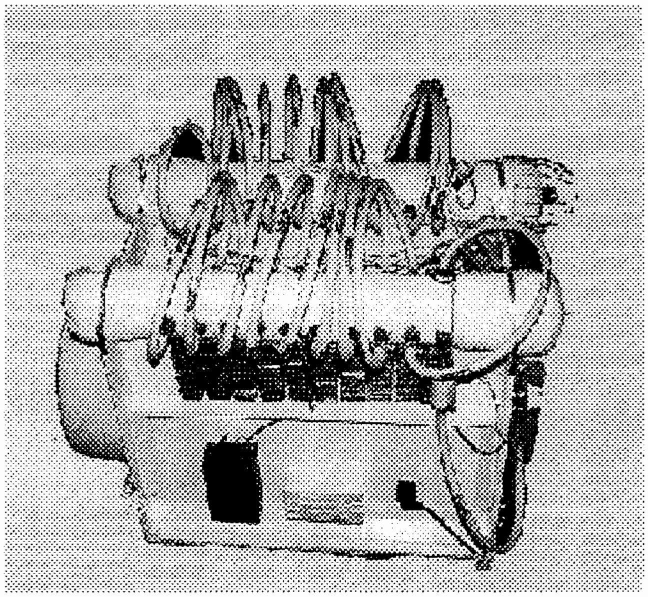
A two stage ECCENTRIC VANE MOTOR has been designed to cope with the volume flow of air at an air pressure of 4.5 MPa. The advantages are that it is compact and lightweight in comparison to a piston cylinder arrangement.

For a power output of 25 kW at 1000 rev/min, two 12 vane rotors are needed. The package is estimated to weigh around 50 kg. Its efficiency should be 95%+ and it is designed to provide propulsion for a payload of four people. Also it must be noted that the POWER PACK is continuously reducing its weight as it uses air from the receiver, giving a power/weight ratio which increases with time.

EXPERIMENTAL TEST RIG

An EXPERIMENTAL TEST RIG has been constructed (left) for examining the performance of the compressed air car power pack. It is a scaled down version of what would be employed on a reasonably sized motor vehicle and, as such, is representative of an actual installation.

Two 12 litre compressed air cylinders pressurized to 30 MPa are used which allow 20 minutes of continuous testing to be carried out before recharge. Two heat exchangers are connected in series and consist of a number of heat pipes. Both heat exchangers are enclosed in a wooden casing through which atmospheric air is enticed by a variable speed fan, and the whole is instrumented to measure pressures, temperatures and flow rates, the data being recorded through a PC based data collection system.



HEAT PIPE HEAT EXCHANGER SYSTEM

If atmospheric air is compressed to high pressures such as 33 MPa it rises in temperature. Before the air is stored in the pressure vessel, it must be cooled to atmospheric temperature, the heat transfer representing an energy loss from the system.

When the air is expanded by passing it through a HEAT EXCHANGER before entry to a motor, some of the energy of compression can be recovered. The HEAT PIPE HEAT EXCHANGER is manufactured in aluminum, will weigh approximately 100 kg and have a facial area of $1.4m^2$, necessitating an atmospheric air velocity of about 2 m/s. It will be of semi circular design (shown right), which has the benefit that it can be compactly fitted around the PRESSURE RECEIVER.

The efficiency level achieved by the TEST RIG is very promising, indicating that a high level of energy recovery is possible. The efficiency determined is that of the heat exchangers and is calculated on the basis of the ratio of the energy actually recovered to the maximum amount of energy that could be recovered. Preliminary testing reveals that efficiencies of 85%+ are achievable!

If the HEAT PIPE HEAT EXCHANGER SYSTEM maintains a high effectiveness across a range of adverse conditions, then the prospects for the introduction of vehicles powered by COMPRESSED AIR are tremendously enhanced.

REGENERATIVE BRAKING

Much energy is lost in the braking process to dissipate the kinetic energy of the car as heat. This energy will be captured and utilized at a rate of 1-2 kW which is representative of CITY DRIVING. The system uses CENTRIFUGALLY VENTILATED DISC BRAKES that draw air in at their centre, pass it over radial vanes and pump it out at the periphery. In braking, the disc brake is heated by friction and cooled by the air, which in turn is shepherded by a shroud around the disc and ducted to the heat exchanger.

ADVANTAGES OF THE COMPRESSED AIR CAR

- ☑ SHORT RECHARGE TIME
- ☑ LONG LIFE
- ☑ HIGH EFFICIENCY
- ☑ LOCALLY NON POLLUTING

HIGH ENERGY CONVERSION EFFICIENCY DEPENDS ON:

- HEAT RECOVERY DURING AIR EXPANSION
- LIGHTWEIGHT MATERIALS
- VARIABLE EXPANSION RATIO ROTARY MOTOR
- RECYCLED HEAT FROM REGENERATIVE BRAKING

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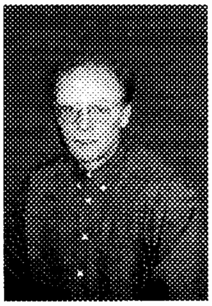

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EMPLOYMENT DETAILS:

1972 - 1977 Contract Research Officer, School of Mechanical Engineering,

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1977 - 1980 Lecturer, Department of Mechanical Engineering,

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1980 - 1986 Senior Lecturer, School of Mechanical Engineering,

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1987 - 1989 Principal Lecturer, School of Mechanical Engineering,

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1990 - 1992 Senior Lecturer, School of Mechanical Engineering,

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1993 - 1997 Director, Division of Design & Mechanical Engineering,

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LECTURING RESPONSIBILITIES AT THE UNIVERSITY OF WESTMINSTER

Primary

Module 3MEA001 - Foundation Mathematics (BEng Course)

Module 3MEA131 - Introductory Mathematics (BEng Course)

Module 3MEA113 - Thermofluid Mechanics (BEng Course)

Module 3MEA184 - Thermofluid Mechanics (HND Course)

Module 3MEA160 - Engineering Principles (ISBM Course)

Module 3NAL284 - Project (HND Course)

Module 3NAL310 - Project (BEng Course)

Module 3ENEM05 - Project (MSc Course)

Module 3ENEM06 - Efficient Energy Management (MSc Course)

Secondary

Module 3MEA203 - Energy Technology (BEng Course)

Module 3MEA303 - Energy Systems Design (BEng Course)

Module 3MEA313 - Energy Management (BEng Course)

Module 3ENEM04 - Energy Systems Technology (MSc Course)

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Visiting Erskine Fellow at the University of Canterbury, Christchurch, New Zealand, 1995.

External Examiner, BSc and BSc Honours Degree course in Computer Aided Engineering at South Bank University, 1994 - 1997.

Member of the Scientific Committee organising the 1st International Conference on 'Energy and the Environment', Limassol, Cyprus, October 12-14, 1997.

Book Reviewer for the International Journal of Ambient Energy and 'Energy World', the Magazine of the Institute of Energy.

RESEARCH AND CONSULTANCY INTERESTS

Energy Management

Human Comfort

Energy Analysis

Heat Pumps

Renewable Energy

Compressed Air Vehicles

RECENT PUBLICATIONS:

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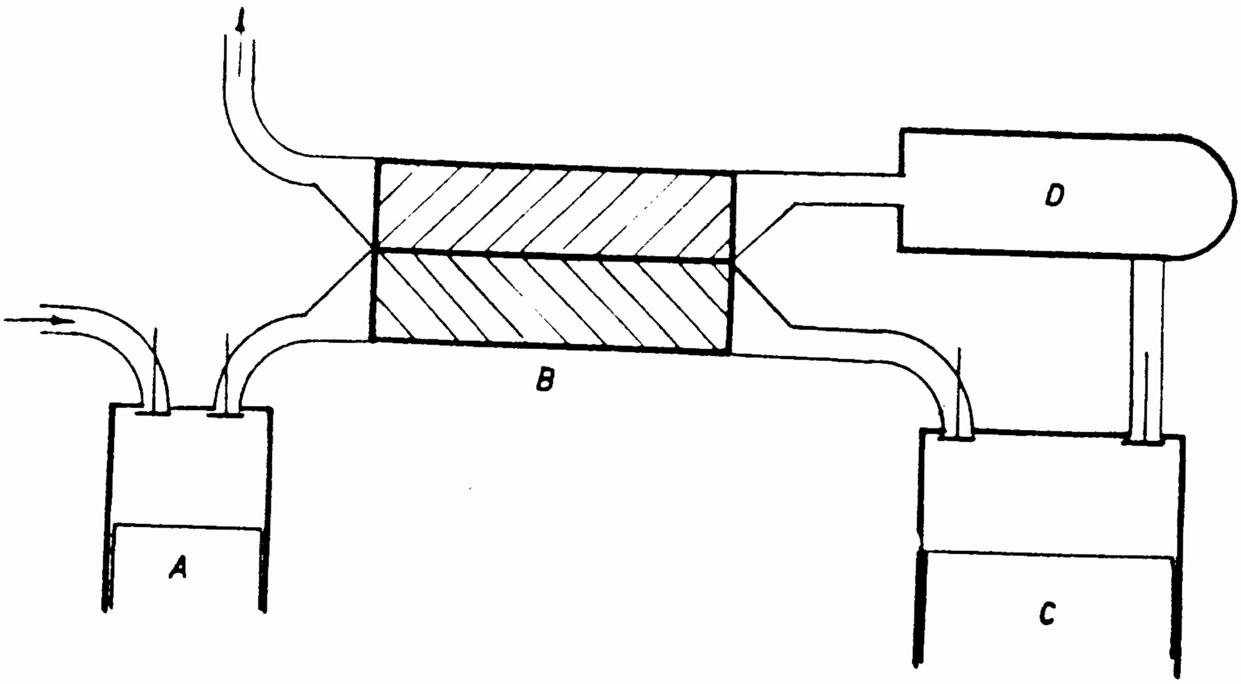
STUDIES FOR A NEW HOT AIR ENGINE

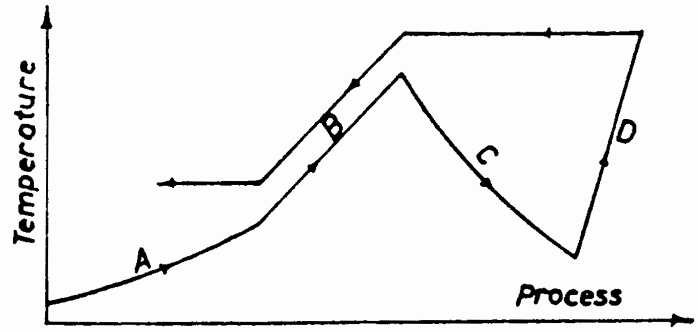
By H. A. Havemann and N. N. Narayan Rao

Received April 15, 1955

Foreword

The thermodynamic analysis of a new Hot Air Engine forms the first account of work initiated and done in the Department of Internal Combustion Engineering under the auspices of the Council of Scientific and Industrial Research (C.S.I.R.) within the " Research Scheme on Hot Air Engines and Their Development," an investigation devoted to fundamental data. It is written in two parts. An account of further work with sub-title "Heat Transfer to Pulsating Air Flowing in a Tube " will be published subsequently as Part III in this Journal.





A: Compressor; B: Heat exchanger; C: expansion motor

D: Combustion chamber

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PART I :

A Principal Thermodynamic Analysis

Summary

This paper deals with the thermodynamic aspects of a new hot air engine cycle which provides for compression of, heat addition to and expansion of, clean air which is used to sustain combustion, with the combustion gases heating externally the compressed air in a heat exchanger. In the first part of this report, the cycle is considered particularly in its simple form without the complications of multistage processes occurring in any of the cylinders. The topics studied include the expansion ratio, cylinder configurations, the concept of necessary and useful expansion, the process of heat addition, and the indices of compression and expansion. The necessary equations are derived and numerical tables are given. In the second part, the effects of introducing multistage units, the air consumption and part-load operation will be examined.

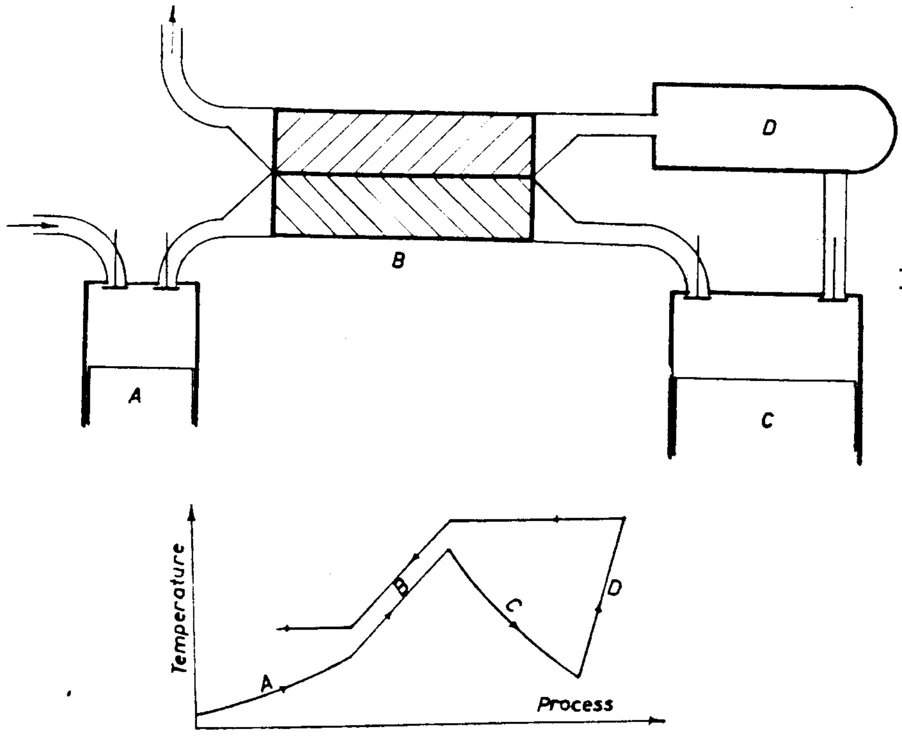
CONTENTS

1. Introduction

The development of a hot air engine, some fundamental aspects of which are examined in this Report, is important where cheap, indigenous, low-grade fuels for power production should be used instead of costly fuels as accepted by the conventional types of internal combustion engines. This is made possible by the introduction of a cycle in which the products of combustion do not come into contact with any of the surfaces of, and between, moving parts of the power unit.

Briefly, the cycle, which by applying terms of gas turbine technology, may be called an exhaust-heated open cycle, can be described as follows.[1*]

Atmospheric air is inducted into a compressor A (Fig. 1) and after being compressed to a suitable pressure, is made to pass through the



A: Compressor; B: Heat exchanger; C: expansion motor
D: Combustion chamber

Fig. 1. The process diagram.

* Numbers refer to the list of references given in Section 8, p. 240.

heat exchanger B where it is heated to a predetermined temperature. The hot and still clean air is now allowed to expand in the expansion motor C where its thermal energy is partly converted into useful mechanical energy. After expansion the air alone or in addition with other air is used as oxygen carrier to burn fuel in a combustion apparatus D, suitably designed for the particular fuel. The products of combustion, mixed with excess air, if necessary, are sent into the heat exchanger B where they give up their heat content as far as is possible, to the incoming compressed air. At the outlet end of the hot side of the heat exchanger, the hot gases would still have some energy left and this may be utilised in various ways depending on the application of the power unit—say for drying moist fuel —and on convenience, or alternatively they may be allowed to escape into the atmosphere.

It will be noticed immediately that parts of the heat exchanger only come into contact with the products of combustion and not any of the surfaces of the moving parts like the piston-cylinder assembly or a turbine wheel in the case of a rotary power plant. Thus, when low-grade, ash-forming fuels are used, the problem of precautions against corrosion and abrasion on moving parts, seizure and similar difficulties are completely eliminated and instead the problem now becomes one of designing for high temperature resisting materials in the heat exchanger where the highest cycle temperature occurs. It may also be pointed out that contamination of the working medium with lubricants will always be restricted to a very small percentage, with no dangerous accumulation being possible.

During the years immediately after the war, another power plant has been developed in Holland by Phillips.[2] This works on the principle of the closed cycle, wherein the same working medium—usually air—is continuously subjected to the thermodynamic processes of the cycle. This results, in the course of time, in the contamination of the working medium with increasing amounts of lubricating oil to the point when the oil vapour medium mixture reaches explosive limits. This difficulty is obviated in the open cycle, described above, as every cycle operates with a fresh portion of the working medium. The cycle under study also eliminates one of the components of the closed cycle, namely the cooler, and thus makes the power unit more suitable for arid zones.

The principle of the exhaust-heated open cycle is not new having been described by many others. It has been mentioned as a possible variation of the gas turbine cycle by Constant,[3] Lysholm,[4] Haverstick[5] and Kreitner and Nettel.[6] Mordell[7] has investigated theoretically the thermodynamics of the cycle and Johnson[8] has described how it could be profitably combined with steam and other turbine cycles to achieve higher efficiencies. Veits and Jacks[9] have investigated the energy characteristics of air turbines. Tyler and McPhail[10] have reported the use of the cycle for the development of a coal-burning gas turbine locomotive. All these researches have been made in connection with the development of high-output gas turbines. No one seems to have specially reported using the cycle for low-output (less than 20 H.P.) reciprocating units. Ricardo and Co. Ltd.,[11] considered the cycle for such a use but seem to be in favour of a small vertical steam engine designed on modern principles. In many backward countries and out-of-the way places, however, the hot air engine has the advantage over the steam engine that it requires no water for its operation. This alone justifies the expenditure of time and energy on the former in preference to the latter, its main problem being to harmonize the quest for good efficiency, *i.e.*, for a high cycle peak temperature with the temperature to which the material in the heat exchanger can safely be subjected.

This part, Part I of the Report, contains a first theoretical analysis of the thermodynamic and related principles of the exhaust-heated open cycle hot air engine as applied to low output reciprocating units.

2. The Simple Cycle

The phrase, "Simple Cycle", is here used to denote the cycle as described at the beginning of the introduction, with single-stage compression and single-stage expansion occurring in two different cylinders and without the complication of multistage processes occurring.

Thermodynamically, the cycle consists of polytropic compression, followed by heat addition substantially at constant pressure and polytropic (in the ideal case, adiabatic) heat rejection.

The ideal efficiency of the cycle may be calculated by assuming that the working fluid is a perfect gas and both the machine components are 100% efficient. Under such circumstances the useful work of the cycle is given by the expression†:

$$L_n = L_e - L_c. \tag{1}$$

The heat supplied to the ideal plant (with a heat exchanger efficiency equal to unity is:

$$Q_i = C_p (T_3 - T_2). \tag{2}$$

The ideal cycle efficiency is therefore

$$\eta_a = \frac{L_n}{JQ_i} = \frac{L_e - L_c}{JC_p (T_3 - T_2)}. \tag{3}$$

Assuming the compression and expansion processes to be isentropic the respective changes in enthalpy will be:

$$L_e = JC_p (T_3 - T_4) \tag{4}$$

and

$$L_c = JC_p (T_2 - T_1), \tag{5}$$

$$\therefore \ \eta_a = \frac{JC_p (T_3 - T_4) - JC_p (T_2 - T_1)}{JC_p (T_3 - T_2)}. \tag{6}$$

† A list of symbols and units is given in Section 9, p. 241.

Since the specific heat C_p is constant for a perfect gas, equation (6) may be simplified into:

$$\eta_a = 1 - \frac{T_4 - T_1}{T_3 - T_2}. \qquad (7)$$

For an ideal process where compression ratio = expansion ratio = E,

$$\frac{T_2}{T_1} = \frac{T_3}{T_4} = E^{n-1}. \qquad (8)$$

Hence, the ideal efficiency η_a is given by the expression

$$\eta_a = 1 - \frac{1}{E^{n-1}}. \qquad (9)$$

This is the same expression as that for the constant pressure Otto cycle. The values of the ideal efficiency at different values of E between 2 and 14 are given in Table I.

The ideal efficiency can be used to obtain the value of the relative efficiency but for purposes of design as well as for comparison with other cycles, the actual or overall thermal efficiency is a more useful concept. This takes into account all the losses which are necessarily involved in the physical design of the power plant.

In a real engine, the actual compression work is given by L_c/η_c and actual expansion work by $\eta_e L_e$. Therefore, the net output per kg of air, reduced by the mechanical losses represented by η_M and the parasite losses represented by Δh is given by the expression

$$L_n = \eta_M \left(\eta_e L_e - \frac{L_c}{\eta_c}\right) - J\Delta h. \qquad (10)$$

If the temperature of the working medium at the end of heat addition (or before the commencement of expansion) is T_3 and the temperature before the beginning of heat addition is T_2, then the heat supplied to the power plant per kg of air is given by

$$L_i = \frac{1}{\eta_B \eta_R} C_p J (T_3 - T_2), \qquad (11)$$

where η_B and η_R are the combustion and heat transfer efficiencies respectively. The overall thermal efficiency can therefore be expressed as:

$$\eta = \frac{L_n}{L_i} = \frac{\eta_M \left(\eta_e L_e - \frac{L_c}{\eta_c}\right) - J\Delta h}{\frac{1}{\eta_B \eta_R} \cdot \bar{C}_p J (T_3 - T_2)}. \qquad (12)$$

In order to simplify this expression for working purposes, the parasite losses, instead of being expressed as Δh may be approximated as a fixed percentage η_L of the net output. Further substituting the following well-known expressions for L_c and L_e namely

$$L_c = \frac{n}{n-1} RT_1 (1 - E^{n-1}) \qquad (13)$$

and

$$L_e = \frac{n}{n-1} RT_3 \left(1 - \frac{1}{E^{n-1}}\right), \qquad (14)$$

equation (12) can be reduced to the form

$$\eta = \eta_L \eta_M \eta_B \eta_R \left(\frac{n}{n-1}\right) \left(\frac{R}{C_p J}\right) \left(\frac{E^{n-1} - 1}{E^{n-1}}\right) \frac{T_3 - \frac{T_1 E^{n-1}}{\eta_e \eta_c}}{T_3 - T_1 E^{n-1}}. \qquad (15)$$

This as well as all the following equations have been evaluated with typical values for the various quantities. Some have been assumed to be constant and some were varied within reasonable ranges. The effect of varying some of the quantities originally assumed to be constant was also ascertained.

The constant values used were: $\eta_L = 0{\cdot}9$, $\eta_M = 0{\cdot}85$, $\eta_B = 0{\cdot}98$ (assuming an oil-fired, gas turbine type combustion chamber), $\eta_R = 0{\cdot}8$, $n = 1{\cdot}3$, $R = 29{\cdot}27$ kg-m/kg-° C, $J = 427$ kg-m/kcal, $\eta_e = 0{\cdot}85$, $\eta_c = 0{\cdot}85$ and $T_1 = 303°$ K (30° C). With substitution of these values and simplification, equation (15) is reduced to

$$\eta = \frac{0{\cdot}1513}{C_p} \frac{E^{0.3} - 1}{E^{0.3}} \frac{T_3 - 419\, E^{0.3}}{T_3 - 303\, E^{0.3}}. \qquad (16)$$

13

The specific heat of air was calculated to the fifth decimal in the relevant temperature ranges.

The change in η with values of E varying from 2 to 14 and T_3 from 325 to 925° C is shown in Table II and plotted in Fig. 2. It can be seen that upto about 600° C the thermal efficiency is a maximum at a compression ratio increasing from 4 to 7 while above that temperature there is no well-defined maximum but the increase in thermal efficiency is negligible for even a considerable increase in compression ratio. As the thermal efficiency is more sensitive to temperature at lower temperatures than at high temperatures, due to the variation of specific heat, the inadvisability of designing for very high temperatures may be deduced. It may also be remembered here that too high a temperature would increase the initial cost of the power plant inordinately due to the necessity of providing high quality steels in the heat exchanger.

Apart from the overall thermal efficiency which represents the effectiveness of utilisation of the fuel, the other important thermodynamic factor to be considered is the specific output which represents the effectiveness of utilisation of the air used in the power plant. The specific output obtained by expressing the net output per kg of air in H.P. per sec is.

$$L_s = \frac{\eta_L \eta_M \eta_C}{76{\cdot}04} \frac{n R}{n-1} \frac{E^{n-1} - 1}{E^{n-1}} \left(T_3 - \frac{T_1 E^{n-1}}{\eta_e \eta_c}\right). \qquad (17)$$

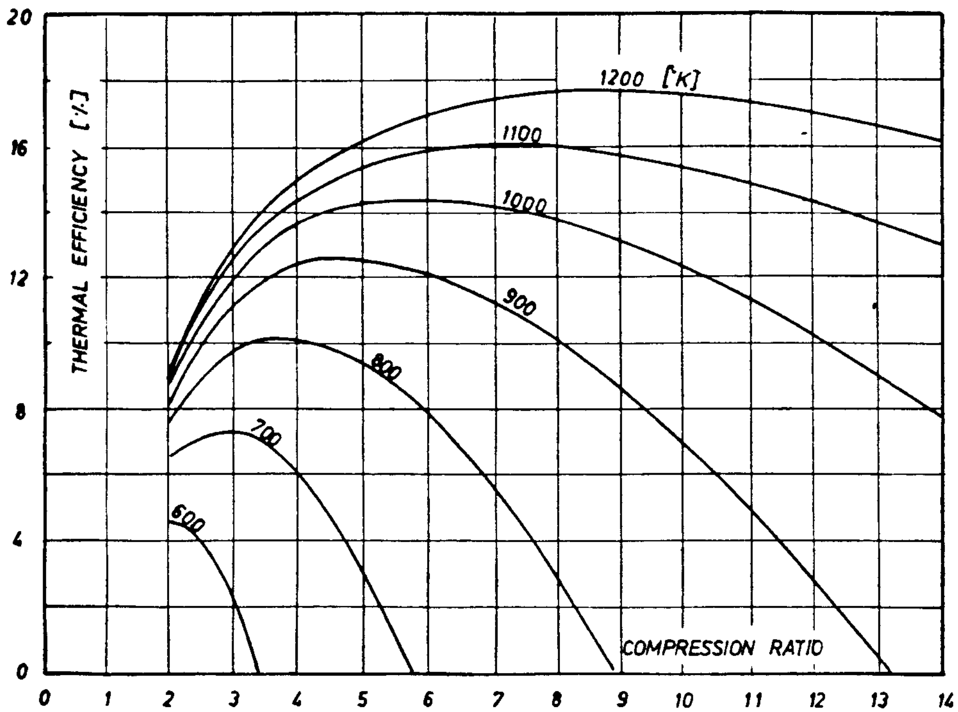


FIG. 2. Overall thermal efficiency.

When the numerical values mentioned above are substituted for the different factors in this equation, it is reduced to the form

$$L_e = 1{\cdot}085 \, \frac{E^{0.3} - 1}{E^{0.3}} \, (T_3 - 419 \, E^{0.3}). \tag{18}$$

The values of L_e for different values of E and T_3 are given in Table III and plotted in Fig. 3. While the specific output is directly proportional to the temperature as may be expected, its maxima, as compared to those of the thermal efficiency, are spread over and shifted to, compression ratios between 5 and 11, increasing with increasing temperatures. Hence the optimum conditions with respect to *both* thermal efficiency and specific output for the simple cycle are given by (*a*) $5 < E < 7$ and (*b*) $T_3 < 800°$ C.

By equating either η or L_e (equations 15, 16, 17 or 18) to zero, $T_{3\,\min.}$ is obtained, which is the temperature necessary to just overcome the losses in the engine

$$T_{3\,\min} = \frac{T_1 \, E^{n-1}}{\eta_e \, \eta_c}\,, \tag{19}$$

or with the numerical values assumed above

$$T_{3\,\min} = 419 E^{n-1}. \tag{20}$$

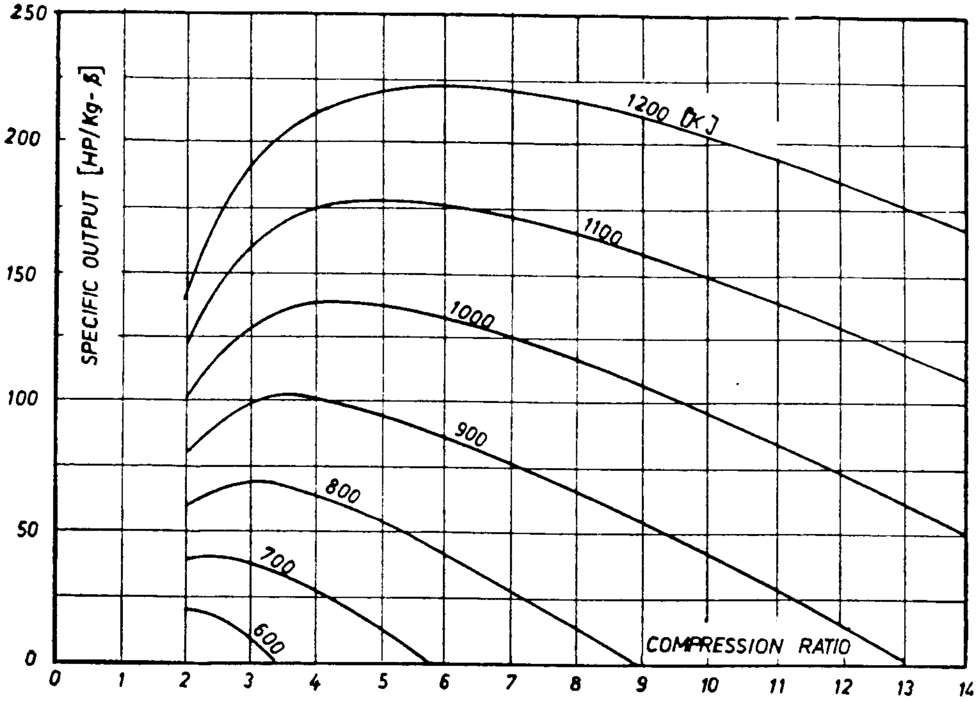


FIG. 3. Specific output.

3. Expansion Ratio

In the analysis it has so far been assumed that the compression and expansion ratios are equal. Matching the two ratios correctly is essential for maintaining the efficiency at a high level. The expansion ratio cannot usually be considerably greater than the compression ratio, as otherwise the pressure at the end of expansion will be too low to overcome the pressure drop across combustion chamber and heat exchanger. This gives the upper limit of the expansion ratio

$$E_{e\,\max} \leqslant E. \tag{21}$$

The lower limit is given by the fact that with low expansion, the output would be less than what is necessary for overcoming the losses in the engine. If, in equations (13) and (14), the value of E is not the same but is E and E_e for compression and expansion respectively, and if these expressions are substituted in equation (12) and the resultant equation is simplified,

$$\eta = \eta_L \eta_M \eta_E \eta_R \, \frac{nR}{n-1} \, \frac{\left[\eta_e T_3 \left(1 - E_e^{\frac{1}{n-1}}\right) - \frac{T_1}{\eta_c}\left(E^{n-1} - 1\right)\right]}{C_p \, J \, (T_3 - T_2)}. \tag{22}$$

The values of η for different values of expansion ratio are given in Table IV. When this is equated to zero and solved for E_e, the minimum value of E_e is

obtained. Hence,

$$E_{e\,\min} \geqslant \left[\frac{1}{1-\dfrac{1}{\eta_e\eta_c}\dfrac{T_1}{T_3}(E^{n-1}-1)}\right]^{\frac{1}{n-1}} . \tag{23}$$

$E_{e\,\min}$ is therefore lowered by increasing T_3 at any compression ratio or by decreasing the compression ratio at any value of T_3.

Within the limits indicated by the expressions (21) and (23) the higher the expansion ratio, the higher will be the thermal efficiency at any given temperature T_3. It is more useful to design a lower compression engine with close values of E and E_e than a higher compression engine with too great a difference between the two. For instance, if $E = 7$ and $E_e = 5$, then $\eta = 6{\cdot}46\%$ at $T_3 = 750°$ C, but if $E = E_e = 5$, $\eta = 13{\cdot}58\%$ at the same temperature.

4. Cylinder Configurations

The calculations so far have been made under the assumption that compression occurs in one cylinder and that expansion occurs in another cylinder of a suitable size. However, some useful deductions follow if the case is considered where the size of the expansion cylinder is arbitrarily chosen. The analysis may be made as follows:—

If p_1, V_{1c} and T_1 are the initial conditions in the compressor and p_2, V_{2c} and T_2 are the final conditions, then

$$G = \frac{p_1\,V_{1c}}{RT_1} = \frac{p_2\,V_{2c}}{RT_2} . \tag{24}$$

If the pressure at the end of heat exchange is p_3, this may be expressed by

$$p_3 = \pi_{23}p_2 = \pi_{23}p_1\,E^n , \tag{25}$$

where π_{23} is a constant which defines the ratio between the pressures at the end and the beginning of the heat exchanger at the moment when the expansion motor inlet valve opens. In the theoretical case, when heat exchange is at constant pressure, $\pi_{23} = 1$; but in the real case, there will be a slight pressure difference between the two due to two reasons: (*a*) the pressure drop across the heat exchanger tubes and (*b*) the effect of constant volume heating which occurs during a short period in each cycle when both the compressor outlet valve and the expansion motor inlet valve remain closed simultaneously. This effect is examined in greater detail in Section 7.

From the value p_3, the pressure at the hot end of the heat exchanger falls when the expansion motor inlet valve opens to the value p_a (say), when the valve closes. p_a is defined by the equation

$$p_a = \pi_{2a}\cdot p_2 = \pi_{2a}\cdot p_1\,E^n , \tag{26}$$

where π_{2a} is a constant.

The drop in pressure from p_3 to p_a may be calculated from

$$\frac{p_3\,V_{HE}}{R\,T_3} = \frac{p_a\,(V_{HE}+V_a)}{RT_a} . \tag{27}$$

At the end of expansion, if the pressure in the working medium is p_4, it may be defined by

$$p_4 = \frac{p_1}{\pi_{41}} . \tag{28}$$

During the expansion process, if p_a, V_a and T_a are the initial conditions and p_4, V_{1e}'' and T_4 are the final conditions, then it follows that

$$mG = \frac{p_a\,V_a}{RT_a} = \frac{p_4\,V_{1e}''}{RT_4} , \tag{29}$$

where m represents the fraction of the air that is used for expansion. V_a may be defined by the expression

$$V_a = \frac{V_{1e}''}{E_e} = V_{1e}''\,C , \tag{30}$$

where C is the cut-off ratio which is here defined as the ratio of the actual volume of hot air admitted into the expansion cylinder to the total volume of the cylinder and is, in effect, the reciprocal of the expansion ratio.

From equations (25), (26), (27) and (30), it is possible to deduce that

$$T_a = \frac{\pi_{2a}}{\pi_{23}}\left(1 + C\,\frac{V_{1e}''}{V_{HE}}\right)(\Delta T + T_1\,E^{n-1}) , \tag{31}$$

where

$$\Delta T = T_3 - T_2 = T_3 - T_1\cdot E^{n-1} . \tag{32}$$

From equations (24) and (29) one obtains

$$\frac{p_a\,V_a}{T_a} = m\,\frac{p_1\,V_{1c}}{RT_1} . \tag{33}$$

From equations (26), (30), (31) and (33), the expression for the cut-off ratio is obtained as

$$C = \frac{1}{\dfrac{\pi_{23}\,E^n}{m\left(\dfrac{\Delta T}{T_1}+E^{n-1}\right)} - \dfrac{V_{1e}''}{V_{1c}\,V_{HE}}} . \tag{34}$$

Again from equations (24) and (29),

$$\frac{p_4\,V_{1e}''}{RT_4} = m\,\frac{p_1\,V_{1c}}{RT_1} , \tag{35}$$

where T_4 is given by the expression

$$T_4 = \frac{T_a}{E_e^{n-1}} = \frac{\pi_{2a}}{\pi_{23}} \left(1 + C \frac{V_{1e}''}{V_{HE}}\right) (\Delta T - T_1 E^{n-1}) C^{n-1}. \qquad (36)$$

From equations (28), (35) and (36), one obtains

$$\pi_{23} = m \frac{V_{1c}}{V_{1e}''} \pi_{41} \pi_{2a} \left(1 + C \frac{V_{1e}''}{V_{HE}}\right) \left(\frac{\Delta T}{T_1} + E^{n-1}\right) C^{n-1}. \qquad (37)$$

If the entire mass of air that is compressed is used for expansion in a single cylinder, then 'm' will be equal to unity. Therefore, equations (34) and (37) may be combined and simplified into the form

$$\frac{V_{1c}}{V_{1e}''} \; \frac{1 + C \frac{V_{1e}''}{V_{HE}}}{C E^n} \left(\frac{\Delta T}{T_1} + E^{n-1}\right) = \pi_{23}. \qquad (38)$$

If, further, π_{23} is equated to unity as an approximation, then

$$\frac{V_{1c}}{V_{1e}''} \; \frac{1 + C \frac{V_{1c}''}{V_{HE}}}{C E^n} \left(\frac{\Delta T}{T_1} + E^{n-1}\right) = 1. \qquad (39)$$

This expression gives the relationship between cut-off ratio and compression ratio for different ratios of compression cylinder volume to expansion cylinder volume. The numerical values are given in Table V. The values shown when the latter ratio is two may be used if production considerations lead to the use of three cylinders for the entire power plant, the first one for compression and the other two for expansion.

On the other hand, compression and expansion might be made to occur in the same cylinder in what might be called the "four-stroke" version of the hot air engine. In such a case the condition is

$$V_{1c} = V_{1e}'', \qquad (40)$$

and equation (39) is simplified into the form

$$\frac{1 + C \frac{V_{1e}''}{V_{HE}}}{C E^n} \left(\frac{\Delta T}{T_1} + E^{n-1}\right) = 1. \qquad (41)$$

The numerical values can be seen in Table V in the column under $C_{1c}/V_{1e}'' = 1$. The value of the cut-off ratio may be obtained by combining equations (34) and (37) to

$$C = \frac{1}{(\pi_{41} \pi_{2a})^{1/n} E}. \qquad (42)$$

These values are listed in Table VI and show that high values of T_3 would lead to high values of the cut-off ratio and consequently to a loss due to high discharge pressure. This is also shown in Fig. 4. If the expansion were to be reasonably complete, the temperature rise ΔT is limited to less than 300° C, giving a value

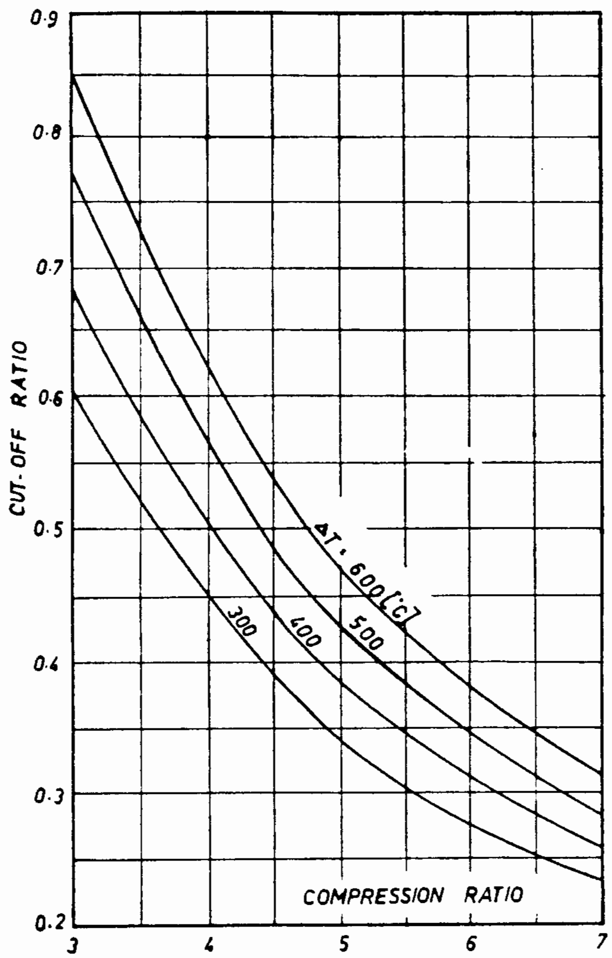


FIG. 4. Cut-off ratio.

of about 450 or 500° C for T_3. At this temperature the efficiency is a maximum at a compression ratio of about 5. At higher temperatures and compression ratios the efficiency drops to uneconomic levels.

The adoption of the single-cylinder unit, while reducing the number of cylinders by half, at the same time decreases—other quantities remaining equal—the mass flow through the power plant and hence the power output, by half.

5. The Concept of Necessary and Useful Expansion

In order to take advantage of higher temperatures and to attain greater flexibility of operation, expansion might be made to take place in two separate cylinders, one designated as the necessary expansion cylinder and the other as the useful expansion cylinder. The former produces just sufficient power for compression and to overcome the losses in the engine while the latter produces the useful power which may be drawn from the shaft and put to work.

Considering first "Necessary Expansion," the condition to be adhered to is

$$G.L_c = m' \, GL_{e \text{ nec.}}, \tag{43}$$

where $m'G$ is the fraction of the air used for "Necessary Expansion". Substituting for L_c and L_e from equations (13) and (14) respectively and solving for m', gives

$$m' = \frac{E^{n-1} - 1}{\left(E^{n-1} + \frac{\Delta T}{T_1}\right)\left(1 - C^{n-1}\right)}. \tag{44}$$

These values are shown in Table VII. It is seen that the part of the air which has to be bled off for necessary expansion decreases with increasing temperature. By combining equations (37) and (44), the relationship between compression and cut-off ratios for necessary expansion is obtained, namely

$$\left(\frac{1 - E^{n-1}}{E^n}\right) \frac{1 + C \frac{V_{1e}''}{V_{HE}}}{C - C^n} = \pi_{23}. \tag{45}$$

Table VIII gives these values when $\pi_{23} = 1$.

The corresponding relationship for useful expansion is obtained by substituting the expression $(1 - m')$ from equation (44) into equation (37) and by simplifying:

$$\left[\left(E^{n-1} + \frac{\Delta T}{T_1}\right) - \left(\frac{E^{n-1} - 1}{1 - C^{n-1}}\right)\right]\left[\frac{1 + C \frac{V_{1e}''}{V_{HE}}}{E^n C}\right] = \pi_{23}. \tag{46}$$

These values are given in Table IX. According to equation (42) the temperatures are limited to about 700° C which is also the limit for easily available heat-resisting steels. At this temperature, m' is about 0·4. In other words, about 60% of the total mass flow produces useful work so that the size of the expansion cylinder may be equal to that of the compression cylinder with very little loss in efficiency but leading to ease of manufacture. In this case, the mass flow and hence power output can be increased by 100% by increasing the number of cylinders by only 50%.

6. The Process of Heat Addition

Heat addition should nominally occur at constant pressure, as otherwise, the compressor will not be able to discharge into the heat exchanger. Nevertheless, it actually takes place partly at constant pressure and partly at constant volume and this effect may be taken advantage of to give a slight increase in the thermal efficiency of the power plant. The effect occurs because the valve timing is usually such that in one part of each cycle, either both the compressor discharge valve and the expansion motor inlet valve remain open or at least one of them is open and heat addition therefore takes place at constant pressure; in the other part of the cycle, both the valves remain closed simultaneously whereby heat addition takes place at constant volume. When the two valves are open during immediately successive intervals (as is necessarily the case when compression and expansion occur in a single cylinder), one gets the maximum possible constant pressure period and the minimum possible constant volume period. The opposite effect is obtained when the overlap of the two opening periods is maximum. In general, constant volume heating may occupy about 60% of each cyclic period.

The problem here is to take advantage of the increased thermal efficiency due to constant volume heating while at the same time overcoming the difficulty of increased resistance to the compressor discharge. This problem can be solved by making the expansion motor inlet period precede the compressor discharge period. Then the complete process in one cycle would be as follows:—

At the moment when the compressor outlet valve opens, the expansion motor inlet valve would be closed and the mean pressure at the inlet of the heat exchanger would be p_2 while the pressure at the outlet of the heat exchanger would be $p_3 = p_2 - \Delta p$ where Δp is the pressure drop along the heat exchanger. When the compressor discharge process is completed, the expansion inlet valve would still be closed and the pressure p_2 would rise at the rate

$$\Delta p_c = p_2 \frac{G}{W_{HE}} \text{ kg/cm}^2 \text{ s}, \tag{47}$$

while the inertia of the air in the heat exchanger would continue the flow and at the same time tend to equalise the pressure. At this time, constant volume heating would commence, increasing the pressure at the rate

$$\Delta p_v = p_2 \frac{H}{W_{HE} C_v} \text{ kg/cm}^2 \text{ s}. \tag{48}$$

Then the expansion motor inlet valve opens and the pressure falls at the rate

$$\Delta p_e = p_3 \frac{G}{W_{HE}} \text{ kg/cm}^2 \text{ s}. \tag{49}$$

The process would be in dynamic equilibrium when

$$c \,.\, \Delta p_c + v \, \Delta p_v = e \, \Delta p_e \tag{50}$$

where c, v and e are the compressor discharge, constant volume heating and expansion motor inlet periods respectively. c, v and e are related by the equation

$$c + v + e = \frac{60}{N} \cdot \tag{51}$$

If it is assumed that

$$\frac{v}{(e + c)} = a \text{ (say)} \tag{52}$$

then, equations (50) and (51) give

$$c = \frac{1 - a\dfrac{\Delta p_v}{\Delta p_e}}{1 + \dfrac{\Delta p_c}{\Delta p_e}} \; \frac{60}{N(a + 1)}, \tag{53}$$

$$e = \frac{1 + a\dfrac{\Delta p_v}{\Delta p_e}}{1 + \dfrac{\Delta p_e}{\Delta p_c}} \; \frac{60}{N(a + 1)}, \tag{54}$$

and

$$v = \frac{60}{N} \cdot \frac{a}{a + 1} \cdot \tag{55}$$

It remains for further examination whether equation (50) is really valid in actual practice. According to the conventional principles of design of a compressor and an expansion motor (or compressed air motor), it may be stated that e is slightly greater than c while v is approximately equal to $2(e + c)$; on the other hand, Δp_e and Δp_c are of the same order of magnitude while Δp_v is comparatively negligible. Hence equation (50) is apparently satisfied but this analysis does not take into account the pressure waves which are generated whenever the valves are operated.

When the compressor discharge valve opens, a high pressure wave is generated and though partially damped by the heat exchanger headers, travels along the heat exchanger tubes at more than sound velocity at the prevailing temperature. This vave is reflected back and forth a number of times depending on the length of the tubes and the magnitude of the constant volume heating period which immediately follows the compressor diacharge period. When the expansion motor inlet valve opens, a low pressure wave is generated and superimposes itself on the high pressure wave. Just sufficient time may be allowed now for this low pressure wave to reach the compressor end of the heat exchanger and then the compressor discharge valve may be opened. A simple calculation shows this time to be of the order of 3 to 4° of crank rotation. In this way the compressor discharge process is improved. Thus this analysis only strengthens the validity of equation (50).

The pressure pulsations described here also have another and possibly beneficial effect on the performance of the heat exchanger. It is proved in a further part of this Report that pulsations increase the heat transfer coefficient in a heat exchanger tube by nearly 30%, if not more, under well-defined conditions of frequency, amplitude and Reynolds Number. Thus, by suitable design, the pulsations can be taken advantage of to decrease the volume of the heat exchanger for any particular power plant. The headers in the heat exchanger can be used to provide just the degree of damping that is necessary to optimise the rate of heat transfer.

7. The Indices of Compression and Expansion

The mean values of the indices of compression as well as expansion have been assumed to be 1·3 in the present calculations. In practice they change with various conditions but their theoretical limits are 1·0 and γ (= 1·4 for air).

During compression, the work input is a minimum with isothermal compression ($n = 1$) and a maximum with adiabatic compression ($n = 1{\cdot}4$); the difference between the two is greater at higher compression ratios. Ideal cooling would abstract all the heat produced by compression and bring the value of n nearer unity but the cooling work increases considerably. On the other hand, compression along the adiabatic line reduces the volumetric efficiency but at the same time, increases the temperature at the end of compression thus allowing for a slight decrease in the duty (or increase in efficiency) of the heat exchanger.

If however, the temperature at the end of compression rises above 200° C. lubrication and sealing difficulties may appear and cooling work increases forbiddingly above a compression ratio of about 10 or 11. The value of n also varies along the p-v curve, being greater at the beginning and smaller at the end than the mean value because of the different quantities of heat flow occurring at different temperature differences. It is further effected by the speed of the compressor, being lower at lower speeds. Similar considerations also apply for the process of expansion during which the work done is a maximum when $n = \gamma$. The value of 1·3 was assumed in the calculations to allow also for the loss due to radiation.

In practice, there is bound to be a variation in the indices and the effect of variation was calculated. By putting n_c and n_e for the indices of compression and expansion in equations (13) and (14) respectively and substituting in equation (12) the following equation is obtained

$$\eta = \eta_L \eta_M \eta_B \eta_K \eta_e \; R \; \frac{\left[\dfrac{n_e}{n_e - 1}\left(1 - \dfrac{1}{E^{n_e - 1}}\right) T_3 - \dfrac{1}{\eta_e \eta_c} \dfrac{n_c}{n_c - 1} T_1 (1 - E^{n_c - 1})\right]}{\bar{C}_p \, J \, (T_3 - T_2)} \cdot \tag{56}$$

Introducing the previously mentioned numerical values and simplifying leads to

$$\eta = 3{\cdot}4946 \; \frac{\left[\dfrac{n_e}{n_e - 1}\left(\dfrac{E^{n_e - 1} - 1}{E^{n_e - 1}}\right) T_3 - 415{\cdot}2 \, \dfrac{n_c}{n_c - 1}(1 - E^{n_c - 1}) T_1\right]}{\bar{C}_p \, J \, (T_3 - T_2)} \cdot \tag{57}$$

Table X shows these numerical values. While variation in the expansion index is not so vital, the thermal efficiency is very sensitive to the compression index. This is presumably so due to the fact that the fall in temperature during expansion is more than twice the rise in temperature during compression.

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9. List of Symbols and Units

Symbol	Significance	Units
c	Compressor discharge period	s
C	Cut-off ratio	..
C_p	Specific heat	kcal/kg-° C
C_p	Mean specific heat at constant pressure	kcal/kg-° C
C_v	Mean specific heat at constant volume	kcal/kg-° C
e	Expansion motor inlet period	s
E	Compression ratio	..
E_r	Expansion ratio	..
G	Mass flow of process air	kg/cycle
H	Rate of heat transfer	kcal/s
J	Mechanical equivalent of heat	kg-m/kcal
L_c	Compressor power input	kg-m/s
L_e	Expansion motor power output	kg-m/s
L_i	Energy input	kg-m/s
L_n	Nett output	kg-m/s
L_s	Specific output	HP/kg-s
m	Fraction of process air used for expansion	..
n	Index of compression or expansion	..
N	Speed of unit	RPM
p_a	Pressure in expansion cylinder when expansion begins	kg/cm², abs
p_1	Compressor suction pressure	kg/cm², abs
p_2	Compressor discharge pressure	kg/cm², abs
p_3	Expansion motor inlet pressure	kg/cm², abs
p_4	Expansion motor discharge pressure	kg/cm², abs
Q_i	Heat input into unit	kcal/s
R	Universal gas constant	kg-m/kg-°C
T_a	Temperature in expansion cylinder when expansion begins	° K
T_1	Compressor suction temperature	° K
T_2	Heat exchanger inlet temperature	° K
T_3	Heat exchanger outlet temperature	° K
$T_{3\ min.}$	Value of T_3 when engine can just overcome its losses	° K
T_4	Temperature at end of expansion	° K
v	Constant volume heating period	s

Symbol	Significance	Units
V_a	Volume in expansion cylinder when expansion begins	m^3
V_{1c}	Volume of compression cylinder	m^3
V_{2c}	Clearance volume in compressor	m^3
V_{1e}	Volume of expansion cylinder	m^3
V_{HE}	Volume of compressed air space in heat exchanger	m^3
W_{HE}	Weight of compressed air filling heat exchanger tubes	kg
α	Ratio of v to $(e+c)$	..
γ	Adiabatic constant	..
Δh	Parasite losses	kcal/s
Δp	Pressure drop in heat exchanger	kg/cm^2
Δp_c	Rate of pressure rise in heat exchanger due to compressor discharge	kg/cm^2–s
Δp_e	Rate of pressure drop in heat exchanger due to expansion motor inlet opening	kg/cm^2–s
Δp_r	Rate of pressure rise in heat exchanger due to constant volume heating	kg/cm^2–s
ΔT	Temperature rise in heat exchanger	°C
η	Overall thermal efficiency	per cent.
η_a	Ideal efficiency	per cent.
η_B	Combustion efficiency	per cent.
η_c	Compression efficiency	per cent.
η_e	Expansion efficiency	per cent.
η_L	Factor used to express $J\Delta h$ as a fraction of L_n	per cent.
η_M	Mechanical efficiency	per cent.
η_R	Heat exchanger efficiency	per cent.
π_{2a}	Ratio of p_a to p_2	..
π_{23}	Ratio of p_3 to p_2	..
π_{41}	Ratio of p_1 to p_4	..

10. List of Tables

11. List of Figures

Table I

Ideal cycle efficiency

E	η_a	E	η_a
2	0·2422	9	0·5848
3	0·3556	10	0·6018
4	0·4256	11	0·6168
5	0·4747	12	0·6298
6	0·5116	13	0·6416
7	0·5408	14	0·6520
8	0·5647		

TABLE II

Overall thermal efficiency

E	T_3 = 600	700	800	900	1000	1100	1200° K
2	4·4592	6·5203	7·5727	8·0248	8·5646	8·8345	8·9930
3	2·1540	7·3958	9·8098	11·1490	11·9797	12·5239	12·8743
4	..	5·9628	10·0916	12·2941	13·6118	14·4380	14·9815
5	..	2·9228	9·2734	12·4331	14·2746	15·4510	16·2705
6	..	..	7·7653	12·0634	14·4605	16·0134	17·0350
7	..	..	5·5811	11·1810	14·2535	16·1506	17·3749
8	..	..	2·8899	9·9371	13·8128	16·0877	17·6142
9	..	..	..	8·5999	13·1315	15·8632	17·6223
10	..	..	..	6·9155	12·3106	15·4352	17·5050
11	..	..	..	5·0469	11·3782	14·9872	17·3186
12	..	..	..	2·8243	10·2808	14·4311	17·0080
13	..	..	..	0·4277	9·0759	13·7739	16·6597
14	..	..	..	..	7·7889	13·0548	16·2655

TABLE III

Specific output

E	T_3 = 600	700	800	900	1000	1100	1200° K
2	18·2	38·4	58·7	79·0	99·2	119·5	139·8
3	7·1	37·5	67·9	98·3	128·7	159·2	189·6
4	..	26·2	63·1	100·0	136·9	173·8	210·7
5	..	11·4	52·9	94·4	136·0	177·2	219·1
6	..	..	40·4	85·5	130·6	175·7	220·7
7	..	..	26·9	74·9	122·8	170·7	218·7
8	..	..	13·0	63·3	113·6	163·9	214·2
9	..	..	..	51·3	103·6	156·0	208·2
10	..	..	..	39·0	93·2	147·2	201·3
11	..	..	..	27·0	82·6	138·2	193·8
12	..	..	..	14·4	71·5	128·4	185·4
13	..	..	..	2·5	60·7	118·9	177·0
14	..	..	..	..	50·0	109·3	168·5

TABLE IV

Overall thermal efficiency for different values of expansion ratio

E_c	E_t	$T_3 = 600$	700	800	900	1000	1100	1200° K
2	2	4·2357	6·1991	7·2049	7·8033	8·1825	8·4392	8·6293
3	2	..	..	..	0·8269	2·4816	3·6321	4·4690
	3	2·1274	7·0777	9·3362	10·6241	11·4248	11·9772	12·3537
4	2	..	..	..	..	..	..	0·6967
	3	..	..	1·4656	4·6744	6·6597	7·9957	8·9505
	4	..	5·2164	9·2599	11·4150	12·7261	13·5924	14·2021
5	3	..	..	..	..	2·5133	4·6199	6·1032
	4	..	..	2·6059	6·5719	8·9414	10·4971	11·5754
	5	..	2·7759	8·7828	11·7952	13·5809	14·7390	15·5250
6	3	..	..	..	..	..	1·2713	3·3197
	4	..	..	..	1·5504	5·1070	7·4094	8·9982
	5	..	..	2·1534	7·1320	9·9948	11·8397	13·0968
	6	..	..	7·3280	11·4094	13·7405	15·2348	16·2377
7	3	..	..	..	..	..	..	0·6596
	4	..	..	..	..	1·3315	4·4177	6·5374
	5	..	..	..	2·4063	6·4632	9·0287	10·7797
	6	..	..	0·7615	6·9583	10·3959	12·5591	14·0308
	7	..	..	5·2980	10·6221	13·5613	15·4021	16·6476
8	4	..	..	..	..	..	1·5082	4·1669
	5	..	..	..	..	2·9678	6·2962	8·5479
	6	..	..	..	2·4292	7·0907	9·9656	11·9053
	7	..	..	..	6·3210	10·4092	12·9190	14·6076
	8	..	..	2·7823	9·5495	13·1620	15·3689	16·8493
9	4	..	..	..	..	..	..	1·8478
	5	..	..	..	..	..	3·5947	6·3572
	6	..	..	..	..	3·7587	7·3975	9·8130
	7	..	..	..	1·8734	7·2277	10·4583	12·5945
	8	..	..	..	5·2970	10·1053	12·9974	14·9019
	9	..	..	0·0013	8·4079	12·7202	15·3046	16·9985
10	5	..	..	..	..	..	0·9146	4·2194
	6	..	..	..	..	0·3947	4·8545	7·7776
	7	..	..	..	..	4·0195	8·0257	10·6414
	8	..	..	..	0·8777	7·0263	10·6563	13·0171
	9	..	..	..	4·1767	9·7586	13·0467	15·1759
	10	..	..	..	6·6303	11·7906	14·8244	16·7814
11	5	..	..	..	..	..	..	2·1000
	6	..	..	..	..	..	2·2994	5·7565
	7	..	..	..	..	0·7448	5·5751	8·6996
	8	..	..	..	..	3·8930	8·2925	11·1410
	9	..	..	..	..	6·7242	10·7617	13·3595
	10	..	..	..	2·3230	8·8443	12·5981	15·0094
	11	..	..	..	4·7950	10·8620	14·3476	16·5813
12	6	..	..	..	..	..	..	3·7418
	7	..	..	..	..	..	3·1042	6·7704
	8	..	..	..	..	0·6399	5·9155	9·2828
	9	..	..	..	..	3·6149	8·4701	11·5657
	10	..	..	..	..	5·8275	10·3700	13·2635
	11	..	..	..	0·3702	7·9355	12·1800	14·8811
	12	..	..	..	2·7339	9·8341	13·8103	16·3386
13	6	..	..	..	..	..	..	1·7616
	7	..	..	..	..	..	0·6544	4·8657
	8	..	..	..	..	..	3·5548	7·4416
	9	..	..	..	..	0·4691	6·1904	9·7819
	10	..	..	..	..	2·7720	8·1505	11·5224
	11	..	..	..	..	4·9660	10·0179	13·1805
	12	..	..	..	..	6·9421	11·6999	14·6740
	13	..	..	..	0·4474	8·6693	13·1700	15·9794
14	7	..	..	..	..	..	..	2·9884
	8	..	..	..	..	..	1·1929	5·6295
	9	..	..	..	..	..	3·9117	8·0294
	10	..	..	..	..	..	5·9337	9·8143
	11	..	..	..	..	1·9432	7·8600	11·5148
	12	..	..	..	..	4·0004	9·5951	13·0464
	13	..	..	..	..	5·7988	11·1116	14·3851
	14	..	..	..	..	7·4674	12·5187	15·6272

TABLE V

Cut-off ratio for different ratios of compression to expansion cylinder volumes

C	E	ΔT = 300	400	500	600° C
0·50	2	1·8124	2·0828	2·3542	2·6246
	3	1·1460	1·3056	1·4658	1·6010
	4	0·8312	0·9412	1·0512	1·1612
	5	0·6466	0·7286	0·8110	0·8932
	6	0·5276	0·5124	0·6574	0·7222
	7	0·4450	0·4582	0·5514	0·6044
	8	0·3840	0·4286	0·4732	0·5178
	9	0·3372	0·3754	0·4138	0·4520
	10	0·3002	0·3336	0·3672	0·4006
	11	0·2704	0·3000	0·3294	0·3590
	12	0·2458	0·2722	0·2986	0·3248
	13	0·2252	0·2488	0·2728	0·2964
	14	0·2076	0·2292	0·2508	0·2724
0·40	2	2·2655	2·3535	2·9427	3·2807
	3	1·4325	1·6320	1·8322	2·0012
	4	1·0390	1·1765	1·3140	1·4515
	5	0·8082	0·9107	1·0137	1·1165
	6	0·6595	0·7405	0·8217	0·9027
	7	0·5562	0·6227	0·6892	0·7555
	8	0·4800	0·5357	0·5915	0·6472
	9	0·4215	0·4692	0·5172	0·5650
	10	0·3752	0·4170	0·4590	0·5007
	11	0·3380	0·3750	0·4117	0·4487
	12	0·3072	0·3402	0·3732	0·4060
	13	0·2815	0·3110	0·3410	0·3705
	14	0·2595	0·2865	0·3135	0·3405
0·30	2	3·0206	3·4713	3·9236	4·3743
	3	1·9100	2·1760	2·4430	2·6683
	4	1·3853	1·5686	1·7520	1·9353
	5	1·0776	1·2143	1·3516	1·4887
	6	0·8793	0·9873	1·0957	1·2037
0·30	7	0·7417	0·8303	0·9190	1·0073
	8	0·6400	0·7143	0·7887	0·8630
	9	0·5620	0·6257	0·6897	0·7533
	10	0·5003	0·5560	0·6120	0·6677
	11	0·4506	0·5000	0·5490	0·5983
	12	0·4097	0·4533	0·4977	0·5413
	13	0·3753	0·4147	0·4880	0·4940
	14	0·3460	0·3820	0·4180	0·4540

TABLE V—*Contd.*

C	E	ΔT = 300	400	500	600° C
0·20	2	4·5310	5·2070	5·8855	6·5615
	3	2·8650	3·2640	3·6645	4·0025
	4	2·0780	2·3530	2·6280	2·9030
	5	1·6165	1·8215	2·0275	2·2330
	6	1·3190	1·4810	1·6435	1·8055
	7	1·1125	1·2455	1·3785	1·5110
	8	0·9600	1·0715	1·1830	1·2945
	9	0·8430	0·9385	1·0345	1·1300
	10	0·7505	0·8340	0·9180	1·0015
	11	0·6145	0·6805	0·7465	0·8120
	12	0·6145	0·6805	0·7465	0·8120
	13	0·5630	0·6220	0·6820	0·7410
	14	0·5190	0·5730	0·6270	0·6810
0·10	2	9·0620	10·4140	11·7710	13·1230
	3	5·7300	6·5280	7·3290	8·0050
	4	4·1560	4·7060	5·2560	5·8060
	5	3·2330	3·6430	4·0550	4·4660
	6	2·6380	2·9620	3·2870	3·6110
	7	2·2250	2·4910	2·7570	3·0220
	8	1·9200	2·1430	2·3660	2·5890
	9	1·6860	1·8770	2·0690	2·2600
	10	1·5010	1·6680	1·8360	2·0030
0·10	11	1·3520	1·5000	1·6470	1·7950
	12	1·2290	1·3610	1·4930	1·6240
	13	1·1260	1·2440	1·3640	1·4820
	14	1·0380	1·1460	1·2540	1·3620
0·05	2	18·1240	20·8280	23·5420	26·2460
	3	11·4600	13·0560	14·6580	16·0100
	4	8·3120	9·4120	10·5120	11·6120
	5	6·4660	7·2860	8·1100	8·9320
	6	5·2760	5·1240	6·5740	7·2220
	7	4·4500	4·5820	5·5140	6·0440
	8	3·8400	4·2860	4·7320	5·1780
	9	3·3720	3·7540	4·1380	4·5200
	10	3·0020	3·3360	3·6720	4·0060
	11	2·7040	3·0000	3·2940	3·5900
	12	2·4580	2·7220	2·9860	3·2480
	13	2·2520	2·4880	2·7280	2·9640
	14	2·0760	2·2920	2·5080	2·7240

Table VI

Cut-off ratio

E	$\pi_{41} = 0 \cdot 5$	0·6	0·7	0·8	0·9	1·0
2	0·869	0·755	0·668	0·604	0·550	0·509
3	0·579	0·503	0·446	0·403	0·367	0·339
4	0·435	0·378	0·334	0·302	0·275	0·255
5	0·348	0·302	0·267	0·242	0·220	0·204
6	0·289	0·252	0·223	0·201	0·184	0·169
7	0·248	0·216	0·191	0·173	0·157	0·146
8	0·217	0·189	0·167	0·151	0·138	0·127
9	0·193	0·168	0·149	0·134	0·122	0·113
10	0·174	0·151	0·134	0·121	0·110	0·102
11	0·158	0·136	0·122	0·110	0·100	0·093
12	0·144	0·126	0·111	0·100	0·092	0·084
13	0·134	0·117	0·103	0·093	0·085	0·078
14	0·124	0·108	0·095	0·086	0·078	0·073

Table VII

Amount of air necessary for expansion

ΔT, °C	E	C = 0·1	0·2	0·3	0·4	0·5	0·6
300	2	0·208	0·271	0·342	0·431	0·551	0·730
	3	0·327	0·426	0·538	0·673	0·869	1·149
	4	0·411	0·535	0·676	0·853	1·091	1·440
	5	0·475	0·619	0·782	0·980	1·159	1·670
	6	0·526	0·649	0·866	1·092	1·400	1·845
	7	0·570	0·742	0·936	1·181	1·521	1·999
	8	0·604	0·786	0·994	1·251	1·603	2·121
	9	0·638	0·833	1·050	1·324	1·700	2·245
	10	0·665	0·868	1·095	1·382	1·770	2·340
	11	0·693	0·904	1·138	1·439	1·840	2·435
	12	0·715	0·932	1·175	1·481	1·899	2·510
	13	0·736	0·958	1·211	1·528	1·955	2·590
	14	0·755	0·984	1·240	1·565	2·003	2·655

Table VII—*Contd.*

ΔT, °C	E	C = 0·1	0·2	0·3	0·4	0·5	0·6
400	2	0·181	0·236	0·298	0·376	0·480	0·636
	3	0·287	0·374	0·472	0·596	0·761	1·008
	4	0·363	0·472	0·597	0·754	0·965	1·272
	5	0·422	0·550	0·684	0·869	1·028	1·481
	6	0·469	0·581	0·771	0·974	1·246	1·641
	7	0·509	0·663	0·837	1·055	1·359	1·789
	8	0·541	0·705	0·890	1·121	1·438	1·899
	9	0·573	0·747	0·944	1·190	1·528	2·015
	10	0·599	0·781	0·986	1·243	1·591	2·105
	11	0·625	0·815	1·025	1·299	1·660	2·195
	12	0·646	0·841	1·061	1·339	1·713	2·265
	13	0·665	0·867	1·095	1·380	1·770	2·340
	14	0·684	0·890	1·125	1·419	1·819	2·403
500	2	0·160	0·208	0·264	0·332	0·425	0·563
	3	0·256	0·334	0·421	0·531	0·680	0·899
	4	0·325	0·423	0·535	0·674	0·863	1·139
	5	0·379	0·494	0·623	0·780	0·924	1·329
	6	0·423	0·527	0·696	0·878	1·122	1·481
	7	0·459	0·599	0·756	0·955	1·229	1·614
	8	0·490	0·637	0·806	1·015	1·300	1·720
	9	0·520	0·678	0·855	1·080	1·382	1·830
	10	0·544	0·710	0·895	1·130	1·448	1·914
	11	0·569	0·742	0·934	1·180	1·510	1·996
	12	0·588	0·767	0·969	1·220	1·562	2·065
	13	0·607	0·791	1·000	1·260	1·615	2·136
	14	0·625	0·815	1·028	1·295	1·661	2·195
600	2	0·144	0·187	0·236	0·298	0·382	0·505
	3	0·234	0·305	0·386	0·479	0·622	0·823
	4	0·294	0·383	0·484	0·610	0·782	1·031
	5	0·344	0·448	0·566	0·710	0·839	1·209
	6	0·385	0·481	0·634	0·799	1·021	1·349
	7	0·419	0·547	0·690	0·870	1·120	1·471
	8	0·448	0·583	0·737	0·930	1·190	1·571
	9	0·476	0·621	0·784	1·013	1·268	1·672
	10	0·498	0·651	0·821	1·038	1·326	1·754
	11	0·521	0·681	0·856	1·082	1·388	1·833
	12	0·541	0·705	0·890	1·121	1·438	1·899
	13	0·509	0·727	0·920	1·160	1·485	1·966
	14	0·576	0·750	0·948	1·191	1·529	2·022

TABLE VIII

Cut-off ratio for necessary expansion

E	C = 0·1	0·2	0·3	0·4	0·5
2	1·8798	1·2245	1·0365	0·9761	0·9989
3	1·8737	1·2206	1·0331	0·9729	0·9957
4	1·7194	1·1201	0·9481	0·8928	0·9137
5	1·5351	1·0000	0·8464	0·7971	0·8158
6	1·3888	0·9047	0·7657	0·7211	0·7380
7	1·2665	0·8251	0·6983	0·6576	0·6731
8	1·1623	0·7572	0·6409	0·6035	0·6177
9	1·0741	0·6997	0·5923	0·5578	0·5708
10	1·0000	0·6514	0·5514	0·5193	0·5314
11	0·9339	0·6084	0·5149	0·4849	0·4963
12	0·8778	0·5718	0·4840	0·4558	0·4665
13	0·8277	0·5392	0·4554	0·4298	0·4398
14	0·7836	0·5104	0·4320	0·4069	0·4164

TABLE IX

Cut-off ratio for useful expansion

C	ΔT = 300	400	500	600° C.
0·1	6·870	7·932	8·983	10·043
0·2	4·374	4·545	5·166	5·725
0·3	2·904	3·303	3·700	4·072
0·4	2·447	2·710	2·903	3·145
0·5	..	2·223	2·488	2·690
0·6	..	..	..	2·226

TABLE X

Overall thermal efficiency for different values of compression and expansion indices

T_3, ° K	n	E	n = 1·20	1·25	1·30	1·35	1·40
600	1·20	2	5·3417	4·2473	3·2728	1·7351	0·2328
		3	6·1516	3·7797	0·6958	..	..
		4	5·2674	0·7215	..	..	..
		5	3·4122	..	..	..	..
		6	0·8806	..	..	..	..
	1·25	2	6·0214	4·9611	4·0231	2·5316	1·0793
		3	6·7592	4·4482	1·4435	..	..
		4	5·9221	1·4817	..	..	..
		5	3·6683	..	..	..	..
	1·30	2	6·4164	5·3760	4·4592	2·9947	1·5714
		3	7·3367	5·0834	2·1540	..	..
		4	6·2765	1·8932	..	..	..
		5	3·7597	..	..	..	..
	1·35	2	7·1430	6·1389	5·2613	3·8462	2·4763
		3	7·9292	5·7353	2·8832	..	..
		4	6·6268	2·3000	..	..	..
		5	3·8329	..	..	..	..
	1·40	2	7·6217	6·6417	5·7898	4·4073	3·0726
		3	8·4387	6·2958	3·5101	..	..
		4	6·8854	2·6002	..	..	..
		5	3·8420	..	..	..	..
800	1·20	2	7·7171	7·1858	6·7433	6·0845	5·4693
		3	10·9440	9·9985	8·9005	7·5637	6·0030
		4	12·5887	11·0368	9·3152	7·0614	4·1407
		5	13·4715	11·5135	9·0059	5·5159	0·4913
		6	13·9147	11·4039	8·6355	2·7403	..
		7	14·0168	10·9133	6·3566	..	..
		8	14·1384	10·1630	4·3263	..	..
		9	14·0379	9·5751	2·2618	..	..
		10	13·3590	8·0623	..	..	..

TABLE X—*Contd.*

T_3, °K	n	E	n = 1·20	1·25	1·30	1·35	1·40
800	1·20	11	12·8947	6·5540	..	..	..
		12	12·3848	5·2572	..	..	..
		13	11·8064	3·5248	..	..	..
		14	11·1232	1·7425	..	..	..
	1·25	2	8·2156	7·6973	7·2678	6·6269	6·0279
		3	11·3652	10·4402	9·3668	8·0596	6·5349
		4	13·0207	11·5003	9·8189	7·6165	4·7612
		5	13·6327	11·6900	9·2030	5·7412	0·9291
		6	13·8488	11·3303	7·8422	2·6393	..
		7	13·7691	10·6298	6·0213	..	..
		8	13·4616	9·6075	3·6469	..	..
		9	13·0202	8·3677	0·7341	..	..
		10	12·4546	6·9664	..	..	..
		11	11·8902	5·3098	..	..	..
		12	11·0665	3·5925	..	..	..
		13	10·3270	1·6177	..	..	..
		14	9·4504	..	..	..	..
	1·30	2	8·5054	7·9947	7·5727	6·9422	6·3526
		3	11·7654	10·8601	9·8098	8·5308	7·0404
		4	13·2547	11·7512	10·0916	7·9171	5·0971
		5	13·6901	11·7530	9·2734	5·8217	0·8543
		6	13·7890	11·2634	7·7653	2·5474	..
		7	13·4440	10·2577	5·5811	..	..
		8	12·9304	8·9887	2·8899	..	..
		9	12·3099	7·5249	..	..	..
		10	11·5332	5·8499	..	..	..
		11	10·6868	3·8194	..	..	..
		12	9·7698	1·9550	..	..	..
		13	8·8145	..	..	..	..
		14	7·7587	..	..	..	..
	1·35	2	9·0384	8·5415	8·1335	7·5220	6·9498
		3	12·1762	11·2909	10·2645	9·0144	7·5591
		4	13·4951	11·9991	10·3610	8·2141	5·4137
		5	13·7361	11·8035	9·3298	5·8861	0·9307
		6	13·6304	11·0862	7·5613	2·3040	..
		7	13·0880	9·8501	5·0990	..	..
		8	12·3800	8·3475	2·1056	..	..
		9	11·5896	6·6704	..	..	..
		10	10·6356	4·7622	..	..	..
		11	9·5881	2·4586	..	..	..
		12	8·4443	0·2812	..	..	..
		13	7·2248	..	..	..	..
		14	6·1011	..	..	..	..
	1·40	2	9·3896	8·9019	8·5030	7·9041	7·3434
		3	12·5293	11·6613	10·6555	9·4302	8·0052
		4	13·6565	12·1822	10·5599	8·4334	5·6741
		5	13·3828	11·8098	9·3368	5·8942	0·9403
		6	13·4090	10·8387	7·2766	1·9642	..
		7	12·7226	9·4319	4·6044	..	..
		8	11·8168	7·6913	1·3030	..	..
		9	10·8231	5·7609	..	..	..
		10	9·6904	3·6168	..	..	..
		11	8·4754	1·0805	..	..	..
		12	7·1584	..	..	..	..
		13	5·8595	..	..	..	..
		14	4·4623	..	..	..	..
1000	1·20	2	8·5065	8·1741	7·8713	7·4656	7·0792
		3	12·4600	11·8886	11·2481	10·4868	9·6704
		4	13·9975	13·9065	13·0093	11·8779	10·5088

TABLE X—*Contd.*

T_3, °K	n	E	n = 1·20	1·25	1·30	1·35	1·40
1000	1·20	5	16·3732	15·3184	14·0747	12·5000	10·5678
		6	17·4571	16·1736	14·5768	12·4800	9·8567
		7	18·2053	16·7204	14·7918	12·2026	8·5459
		8	18·7721	17·0427	14·7736	11·5444	6·7138
		9	19·4471	17·4768	14·7946	10·8807	4·6403
		10	19·5371	17·2647	14·0934	9·2879	0·9253
		11	19·6457	17·1186	13·5684	7·8002	..
		12	19·8401	17·1079	12·9162	6·0626	..
		13	19·9287	16·8214	12·1409	3·9301	..
		14	19·9313	16·5823	11·2905	1·4133	..
	1·25	2	8·9308	8·6065	8·3097	7·9142	7·5359
		3	12·8110	12·2508	11·6232	10·8764	10·0777
		4	15·1157	14·2771	13·4002	12·2924	10·9500
		5	16·5031	15·4562	14·2220	12·6590	10·7425
		6	17·4047	16·1175	14·5159	12·4129	9·7808
		7	18·0109	16·5092	14·5590	11·9402	8·2395
		8	18·4021	16·6374	14·3191	11·0214	6·0790
		9	18·6679	16·6144	13·8152	9·7226	3·1773
		10	18·8508	16·4987	13·2103	8·2160	..
		11	18·8941	16·2661	12·5719	6·5560	..
		12	18·8639	15·9922	11·5876	4·3597	..
		13	18·8433	15·5711	..	1·9241	..
		14	18·7175	15·1691	..	..	..
	1·30	2	9·1774	8·8578	8·5646	8·1750	7·8014
		3	13·1445	12·5951	11·9797	11·2467	10·4648
		4	15·3070	14·4777	13·6118	12·5168	11·1889
		5	16·5494	15·5054	14·2746	12·7158	10·8049
		6	17·3571	16·0665	14·4605	12·3519	9·7118
		7	17·7558	16·2320	14·2535	11·5958	7·8373
1000	1·30	8	17·9909	16·1859	13·8128	10·4347	5·3718
		9	18·1239	16·0124	13·1315	8·9142	2·1560
		10	18·1516	15·6894	12·3106	7·1238	..
		11	17·9936	15·2449	11·3782	5·0656	..
		12	17·9037	14·8948	10·2808	2·6847	..
		13	17·7336	14·2929	9·0759	..	..
		14	17·4900	13·7400	7·7889	..	..
	1·35	2	9·6309	9·3200	9·0333	8·6546	8·2897
		3	13·4808	12·9483	12·3455	11·6266	10·8620
		4	15·4960	14·6759	13·8210	12·7385	11·4249
		5	16·5866	15·5447	14·3167	12·7613	10·8548
		6	17·2310	15·9314	14·3137	12·1902	9·5289
		7	17·4763	15·9284	13·9188	11·2186	7·3969
		8	17·5644	15·7181	13·2882	9·8350	4·6391
		9	17·5724	15·4020	12·4383	8·0945	1·1205
		10	17·4605	14·9579	11·4341	6·0599	..
		11	17·1715	14·3135	10·2883	3·7048	..
		12	16·9221	13·7730	8·9450	0·9725	..
		13	16·5672	12·9494	7·4473	..	..
		14	16·2871	12·3396	6·0636	..	..
	1·40	2	9·9298	9·6246	9·3422	8·9706	8·6114
		3	13·7811	13·2521	12·6601	11·9533	11·2036
		4	15·6355	14·8222	13·9754	12·9022	11·5991
		5	16·5912	15·5497	14·3219	12·7669	10·8610
		6	17·0548	15·7427	14·1087	11·9645	9·2736
		7	17·1896	15·6169	13·5754	10·8315	6·9449
		8	17·1279	15·2393	12·7514	9·2173	3·8894
		9	16·9854	14·7524	11·7005	7·2222	0·0185
		10	16·7532	14·5721	10·5111	4·9396	..

TABLE X—*Contd.*

T_3, ° K	*n*	E	*n* = 1·20	1·25	1·30	1·35	1·40
1000	1·40	11	16·3389	13·3682	9·1846	2·3268	..
		12	15·9698	12·6847	7·6491	..	..
		13	15·5655	11·7956	6·0486	..	..
		14	15·0979	10·9551	4·3580	..	..
1200	1·20	2	8·8336	8·5606	8·3728	8·0926	7·8065
		3	13·0967	12·7033	12·2337	11·7289	11·2059
		4	15·7081	15·1260	14·4646	13·7655	12·9507
		5	17·6157	16·8713	16·1009	15·1042	13·9861
		6	18·9423	18·1217	17·1322	15·8748	14·4312
		7	19·9969	18·9785	17·8167	16·3950	14·5211
		8	20·7445	19·6660	18·3929	16·6648	14·4350
		9	21·6733	20·4268	18·9528	16·9481	14·2668
		10	21·9631	20·6393	18·9098	16·5654	13·3031
		11	22·3789	20·8093	19·0201	16·3205	12·4743
		12	22·7375	21·1724	19·0173	16·0387	11·4474
		13	23·0493	21·2643	18·9621	15·6296	10·1456
		14	23·2863	21·4455	18·8530	15·0747	8·6132
	1·25	2	9·2161	8·7947	8·7650	8·4921	8·2109
		3	13·4097	13·0240	12·5622	12·0670	11·5556
		4	16·0208	15·4501	14·8000	14·1159	13·3181
		5	17·7300	16·9902	16·2259	15·2359	14·1265
		6	18·8966	18·0737	17·0813	15·8204	14·3724
		7	19·9865	18·9674	17·8047	16·3821	14·5068
		8	20·4262	19·3262	18·0246	16·2607	13·9791
		9	21·0043	19·7098	18·1692	16·0758	13·2652
		10	21·3790	20·0073	18·2139	15·7802	12·3796
		11	22·1094	20·5167	18·6939	15·9478	11·4099
		12	21·9194	20·2696	18·0044	14·8628	9·9941
		13	22·1376	20·2609	17·8237	14·2837	8·4427
		14	22·2710	20·3185	17·5665	13·5277	6·5984
	1·30	2	9·4384	9·1889	8·9930	8·7243	8·4459
		3	13·7071	13·3288	12·8743	12·3883	11·8879
		4	16·1901	15·6255	14·9815	14·3056	13·5170
		5	17·7708	17·0327	16·2705	15·2829	14·1767
		6	18·8551	18·0300	17·0350	15·7709	14·3188
		7	19·6062	18·5656	17·3749	15·9155	13·9927
		8	20·0716	18·9478	17·6142	15·8105	13·4711
		9	20·5373	19·2093	17·6223	15·4670	12·5660
		10	20·7839	19·3634	17·5050	14·9802	11·4398
		11	20·9736	19·2825	17·3186	14·3759	10·1347
		12	21·1039	19·3815	17·0080	13·7062	8·5645
		13	21·2054	19·2350	16·6597	12·9077	6·7017
		14	21·2442	19·1788	16·2655	11·9633	4·5609
	1·35	2	9·8472	9·6030	9·4122	9·1514	8·8782
		3	14·0124	13·6415	13·1946	12·7180	12·2289
		4	16·3575	15·7989	15·1610	14·4931	13·7136
		5	17·8034	17·0667	16·3062	15·3205	14·2168
		6	18·7451	17·9143	16·9122	15·6399	14·1770
		7	19·3633	18·3089	17·1003	15·6174	13·6643
		8	19·7042	18·5556	17·1891	15·3440	12·9448
		9	20·0639	18·7018	17·0677	14·8496	11·8571
		10	20·2041	18·7361	16·8143	14·2008	10·5242
		11	20·2743	18·5228	16·4718	13·4083	8·9705
		12	20·2758	18·4738	15·9896	12·5239	7·1033
		13	20·2256	18·1567	15·4363	11·4613	4·8718
		14	20·2380	18·0620	14·9907	10·4303	2·5643
	1·40	2	10·1166	9·8758	9·6885	9·4328	9·1630
		3	14·2748	13·9105	13·4700	13·0015	12·5221
		4	16·4810	15·9269	15·2934	14·6315	13·8587
		5	17·8075	17·0709	16·3107	15·3252	14·2218

TABLE X—*Contd.*

T_3, °K	n	E	n=1·20	1·25	1·30	1·35	1·40
1200	1·40	6	18·5914	17·7526	16·7408	15·4570	13·9790
		7	19·1141	18·0455	16·8185	15·3116	13·3273
		8	19·3283	18·1543	16·7540	14·8667	12·4063
		9	19·5599	18·1617	16·4775	14·1926	11·1026
		10	19·5662	18·0756	16·0871	13·3802	9·5601
		11	19·5662	17·7533	15·6144	12·4283	7·7915
		12	19·4725	17·5931	15·0015	11·3769	5·6856
		13	19·3842	17·2307	14·3856	10·2192	3·3002
		14	19·2433	16·9579	13·7304	8·9148	0·5905

STUDIES FOR A NEW HOT AIR ENGINE

BY H. A. HAVEMANN AND N. N. NARAYAN RAO

PART II :

A Further Thermodynamic Analysis

SUMMARY

The second part of the Thermodynamic Analysis for a New Hot Air Engine is devoted to an investigation into the effects and advisability of subdividing compression and expansion into stages, with cooling or heating applied respectively between two stages. The requirements of combustion air are considered and the behaviour of the plant at part-load for several governing methods are discussed. In the conclusions the broad principles for a design as emanating from the analytical treatment are worked out, and suggestions are made for a suitable design of simple manufacture and operation. After an assessment of the advantages and limitations of the hot air engine suggestions are put forward for further investigating fundamental data on which a future design can be based which would ensure reasonably favourable operation.

CONTENTS

1. INTRODUCTION

In Part I of this paper, the hot air engine cycle was described and some of its simple thermodynamic cycle characteristics were examined. In this part further and more complex characteristics will be studied, namely, the effect of using multistage units for compression and expansion, the distribution of air in the cycle and part-load operations, and from these conclusions will be drawn for a suitable construction of a hot air engine power plant as can be derived from theoretical considerations.

2. THE MULTISTAGE CYCLE

A glance at Tables II and III in Part I of this Report shows that the order of efficiencies and specific outputs to be expected from the hot air engine is very low. One of the possible methods of improving both efficiency and specific output is the introduction of multistage processes either for compression or for expansion or both.

As a first step, the effect is studied of converting the compression process into a two-stage process and retaining the expansion process as a single-stage process. Under assumed conditions of perfect intercooling and ideal intercooler pressure, the compression work is given by

$$L_c = \frac{2n}{n-1} RT_1 \left(1 - E^{\frac{n-1}{2}}\right) \tag{58}$$

Introducing this expression into the expression for the efficiency

$$\eta = \frac{\eta_M \left(\eta_e L_e - \frac{L_c}{\eta_c}\right) - J\Delta h}{\frac{1}{\eta_B \eta_R} \cdot C_p J (T_3 - T_2')} \tag{12}$$

and simplifying, one obtains

$$\eta = \eta_L \eta_M \eta_B \eta_R \frac{nR}{n-1} \frac{\left[\eta_e T_3 \left(1 - \frac{1}{E^{n-1}}\right) - \frac{2T_1}{\eta_c}\left(E^{\frac{n-1}{2}} - 1\right)\right]}{C_p J (T_3 - T_2')} . \tag{59}$$

In terms of the numerical values given in para 2 of Part I, this becomes

$$\eta = 0 \cdot 1516 \frac{\left[T_3 \left(\frac{E^{0.3} - 1}{E^{0.3}}\right) - 830 \cdot 7 (E^{0.15} - 1)\right]}{C_p (T_3 - 300 E^{0.15})} . \tag{60}$$

Similarly, the specific output is given by

$$L_s = \frac{\eta_L \eta_M}{76 \cdot 04} \frac{nR}{n-1} \left[T_3 \left(\frac{E^{n-1} - 1}{E^{n-1}}\right) - \frac{2T_1}{\eta_e \eta_c} \left(E^{\frac{n-1}{2}} - 1\right)\right] \cdot \tag{61}$$

In terms of the numerical values, this becomes

$$L_s = 1 \cdot 084 \left[T_3 \left(\frac{E^{0.3} - 1}{E^{0.3}}\right) - 830 \cdot 7 (E^{0.15} - 1)\right] \cdot \tag{62}$$

From equations (59) or (61), the expression for $T_{3\,min}$ can be derived

$$T_{3\,min} = 830 \cdot 7 \frac{E^{0.3} (E^{0.15} - 1)}{(E^{0.3} - 1)} . \tag{63}$$

Tables XI and XII give the numerical values of η and L_s respectively, for two-stage compression. The efficiency actually falls below that of the pure single-stage

process in the region: $E < 8$ and $T_3 > 750°$ C. Beyond these values, an increase in η can be noticed, of a magnitude equivalent to raising T_3 by about 100° C. The specific output is $1\frac{1}{4}$ to $1\frac{1}{2}$ times higher at all temperatures and compression ratios. $T_{3\,\min}$ is reduced by about 10% by the multiplying factor

$$2\,\frac{E^{\frac{n-1}{2}}-1}{E^{n-1}-1}\,. \tag{64}$$

If the two-stage compression with perfect intercooling is combined with the two-stage expansion, then

$$L_e = \frac{n}{n-1}\,RT_3\left(1+\frac{T_3'}{T_3}\right)\left[1-\left(\frac{p_4}{p_3}\right)^{\frac{n-1}{2n}}\right], \tag{65}$$

where T_3'/T_3 is the reheat factor, defined by the equation

$$T_3 < T_3'\,E^{\frac{n-1}{2}}. \tag{66}$$

Combining equations (12), (58) and (65). the overall thermal efficiency is obtained as

$$\eta = \eta_L\,\eta_M\,\eta_B\,\eta_R\,\frac{nR}{n-1}\,\frac{\left\{\eta_c\,T_3\left(1+\frac{T_3'}{T_3}\right)\left[1-\left(\frac{p_4}{p_3}\right)^{\frac{n-1}{2n}}\right]-\frac{2T_1}{\eta_c}\left[\left[\frac{p_2}{p_1}\right]^{\frac{n-1}{2n}}-1\right]\right\}}{\bar{C}_p\,J\left(T_3-T_1\,E^{\frac{n-1}{2}}\right)} \tag{67}$$

or

$$\eta = \frac{15\cdot16\,(E^{0.15}-1)}{\bar{C}_p(T_3-300\,E^{0.15})}\left[\frac{T_3}{E^{0.15}}\left(1+\frac{T_3'}{T_3}\right)-600\right]. \tag{68}$$

In a similar manner, the expression for the specific output is obtained, namely

$$L_s = \frac{\eta_L\,\eta_M}{76\cdot04}\left[\eta_c\,\frac{n}{n-1}\,RT_3\left(1+\frac{T_3'}{T_3}\right)\left(1-\frac{1}{E^{\frac{n-1}{2}}}\right)+\right.$$

$$\left.+\frac{1}{\eta_c}\,\frac{2n}{n-1}\,RT_1\left(1-E^{\frac{n-1}{2}}\right)\right] \tag{69}$$

or

$$L_s = 1\cdot084\,(E^{0.15}-1)\left[\frac{T_3}{E^{0.15}}\left(1+\frac{T_3'}{T_3}\right)-830\cdot7\right]. \tag{70}$$

By equating equations (68) or (70) to zero, $T_{3\,\min}$ is obtained:

$$T_{3\,\min} = \frac{830\cdot7\,E^{0.15}}{1+\frac{T_3'}{T_3}}. \tag{71}$$

Tables XIII and XIV give the numerical values of η and L_s respectively. An apparent increase in thermal efficiency of between 5 to 25% occurs over that of the single-stage process, the advantage increasing with increasing compression ratios. Nevertheless, this is more than offset, particularly for compression ratios of less than about 7, if the energy spent in reheating is taken into consideration, the decrease due to this being of the order of 5 to 12%. The importance of reheat is seen from the fact that the thermal efficiency falls below that of the single-stage process if the reheat is less than 75% of perfect reheat. The specific output is not appreciably increased over the previous case and remains at between $1\frac{1}{4}$ and $1\frac{1}{2}$ times that of the single-stage process. $T_{3\,\min}$ is decreased by about 20%.

If three-stage compression and expansion were to be used, then the respective compression and expansion work is given by the expressions

$$L_c = \frac{3n}{n-1}\,RT_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{3n}}-1\right] \tag{72}$$

and

$$L_e = \frac{n}{n-1}\,RT_1\left(1+\frac{T_3'}{T_3}+\frac{T_3''}{T_3}\right)\left[1-\left(\frac{p_4}{p_3}\right)^{\frac{n-1}{3n}}\right]. \tag{73}$$

Combining these with equation (12) gives

$$\eta = \eta_L\eta_M\eta_B\eta_R\,\frac{nR}{n-1}\,\frac{\left\{\eta_c\,T_3\left(1+\frac{T_3'}{T_3}+\frac{T_3''}{T_3}\right)\left[1-\left(\frac{p_4}{p_3}\right)^{\frac{n-1}{3n}}\right]-\frac{3T_1}{\eta_c}\left[\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{3n}}-1\right]\right\}}{\bar{C}_p\,J\left(T_3-T_1\,E^{\frac{n-1}{3}}\right)} \tag{74}$$

or

$$\eta = \frac{15\cdot16\,(E^{0.1}-1)}{C_p(T_3-300\,E^{0.1})}\left[\frac{T_3}{E^{0.1}}\left(1+\frac{T_3'}{T_3}+\frac{T_3''}{T_3}\right)-900\right], \tag{75}$$

where

$$T_3 < T_3'\,E^{\frac{n-1}{3}} \tag{76}$$

and

$$T_3' < T_3''\,E^{\frac{n-1}{3}}. \tag{77}$$

Similarly, the specific output is given by

$$L_s = \frac{\eta_L\,\eta_M}{76\cdot04}\,\frac{nR}{n-1}\left(E^{\frac{n-1}{3}}-1\right)\left[\frac{\eta_c\,T_3}{E^{\frac{n-1}{3}}}\left(1+\frac{T_3'}{T_3}+\frac{T_3''}{T_3}\right)-\frac{3T_1}{\eta_c}\right] \tag{78}$$

or

$$L_s = 1\cdot084\,(E^{0.1}-1)\left[\frac{T_3}{E^{0.1}}\left(1+\frac{T_3'}{T_3}+\frac{T_3''}{T_3}\right)-1246\right]. \tag{79}$$

From equations (74) or (78), $T_{3\,min}$ is obtained as

$$T_{3\,min} = \frac{3\,T_1}{\eta_c} \; \frac{E^{\frac{n-1}{3}}}{1 + \frac{T_3'}{T_3} + \frac{T_3''}{T_3}}. \tag{80}$$

The numerical values of η and L_s are given in Tables XV and XVI respectively.

In all the multistage systems considered here, the gain in thermal efficiency is practically negligible, if the inevitably imperfect intercooling and reheat are taken into consideration, except for very high compression ratios working at low temperatures. Though the specific output is increased by about 25%, in view of the greater weight of the power plant, multistage units do not seem to be promising propositions for the small output reciprocating hot air engine.

3. Air Consumption

Under particular conditions of compression ratio, temperature and component efficiencies, the heat input into the power plant [from equation (12)] will be

$$Q_1 = \frac{C_p}{\eta_B\,\eta_R} \; (T_3 - T_2) \text{ kcal/kg process air.} \tag{81}$$

If B kcal/kg were the calorific value of the fuel used, the amount of fuel necessary will be given by the expression

$$F = \frac{\bar{C}_p\,(T_3 - T_2)}{B\,\eta_B\,\eta_R} \text{ kg fuel/kg process air.} \tag{82}$$

For any fuel, solid, liquid or gaseous, whose calorific value is in the region $3400 < B < 11{,}350$ kcal/kg, the amount of air necessary for the release of 10^6 kcal is 135·5 kg. Hence the amount of air necessary to burn the fuel in equation (82) is given by

$$A = \frac{a\,\bar{C}_p\,(T_3 - T_2)}{738\,\eta_B\,\eta_R} \text{ kg combustion air/kg process air,} \tag{83}$$

where a is the excess air factor. This can be written as

$$A = \frac{a}{738}\,Q_1. \tag{84}$$

Therefore when

$$Q_1 = \frac{738}{a}, \tag{85}$$

the process air is just sufficient to burn the necessary fuel. When

$$Q_1 > 738/a, \tag{86}$$

the process air is insufficient for combustion. When

$$Q_1 < 738/a, \tag{87}$$

some of the process air is left over after combustion. This is the case in the usual range of values as can be seen from Table XVII, calculated from equation (83). The extra air can then be used for secondary purposes such as drying the fuel, running an exhaust gas turbine or a secondary expansion motor, or the like.

However, favourable conditions in the heat exchanger require that the water equivalent on both the hot and cold sides be approximately equal. Since the specific heats of hot air and combustion products are nearly the same, the mass flows on both sides should also be the same. Hence as much of the extra air as possible should be led into the heat exchanger, bypassing the combustion chamber. This will of course cool the combustion products and hence the air should be led into the hot side of the combustion chamber at a point where the temperature of the hot gases is equal to the temperature of the air.

4. Part-Load Operations

A hot air engine may be run at part-load by reducing either the mass flow or the temperature at the inlet to the expansion motor, or both simultaneously. The mass flow may be reduced by throttling the compressor inlet or by bleeding off the compressed air after the heat exchanger and using it for secondary purposes. The temperature may be reduced by reducing the fuel feed or by increasing the proportion of the secondary air to the combustion chamber. Part-load conditions can also be obtained by increasing the cut-off ratio or by changing the speed but in most applications, these will be found impracticable.

In the first instance, it may be assumed that the full load mass flow M_1 is reduced to M_1/d_1 by throttling the compressor inlet and that the full load temperature T_3 is reduced to T_3/d_2 at some part-load. This ensures that the same weight of air is passing through all the components of the power plant. Assuming that the compressor and expansion motor are not coupled the expansion motor output is reduced to $\eta_e L_e/d_1 d_2$ and the compressor input is reduced to $L_c/\eta_c d_1$, while the heat input into the engine is reduced to

$$\frac{1}{\eta_B\,\eta_R}\; C_p \cdot J\left(\frac{T_3}{d_2} - T_2\right). \tag{88}$$

Hence η_d the overall thermal efficiency at part-load equivalent to $1/d$ full load (where $d = d_1 d_2$) is given by

$$\eta_d = \frac{\eta_L\,\eta_M\left(\frac{\eta_e\,L_e}{d_1\,d_2} - \frac{L_c}{\eta_c\,d_1}\right)}{\frac{1}{\eta_B\,\eta_R}\,\bar{C}_p\,J\left(\frac{T_3}{d_2} - T_2\right)}. \tag{89}$$

When $d = d_1 = d_2 = 1$, this expression reduces itself to the expression (12) for the full load thermal efficiency. By substituting the usual expressions for L_c and L_e respectively and the relevant numerical values, the equation (89) can be simplified to the form

$$\eta_d = \frac{1}{d_1} \cdot \frac{0{\cdot}1513}{C_p} \frac{E^{0.3} - 1}{E^{0.3}} \frac{T_3 - 420\, d_2\, E^{0.3}}{T_3 - 303\, d_2\, E^{0.3}} . \tag{90}$$

Table XVIII gives these numerical values. By differentiating η_d with respect to $1/d_2$ and equating to zero, it is found that η_d is a maximum when

$$\frac{1}{d_2} = \frac{642\, E^{0.3}}{T_3} . \tag{91}$$

This shows that if part-load conditions are obtained by reducing the temperature T_3 alone and keeping the mass flow constant, then the cut-off ratio should be correspondingly increased. On the other hand, η_d is directly proportional to $1/d_1$. Hence if the mass flow alone is reduced, the efficiency falls proportionately but can be increased again by decreasing the temperature.

The specific output under part-load conditions can be easily obtained as

$$Ls_d = \frac{1}{d} \frac{\eta_L\, \eta_M\, R}{76{\cdot}04} \frac{n}{n-1} (E^{n-1} - 1) \left(\frac{\eta_c\, T_3}{E^{n-1}\, d_2} - \frac{T_1}{\eta_c} \right) \tag{92}$$

or

$$Ls_d = \frac{1{\cdot}083}{d_1\, d_2} \frac{E^{0.3} - 1}{E^{0.3}} (T_3 - 420\, d_2\, E^{0.3}). \tag{93}$$

5. Conclusions

The following broad generalisations can be made regarding the exhaust heated Hot Air Engine cycle applied to low-output reciprocating units, on the basis of the calculations and analysis presented in the two parts of this Report:

(*a*) In its simple form both the efficiency and specific output are rather low, the optimum conditions being a compression ratio in the range of 5 to 7 and a value of $T_3 < 800°$ C.

(*b*) The efficiency can be increased by introducing two or three stage compression and expansion processes. Taking into account the inevitably imperfect reheat and intercooling processes, the increase in efficiency is appreciable only at low values of T_3 and at high compression ratios. The increase in initial cost due to the necessity of reheaters, intercoolers and multiple cylinders of different sizes, as well as due to the higher pressures for which the heat exchanger and the piping system has to be designed, makes the multistage hot air engine cycle *for low output engines*, largely a matter of theoretical interest only.

(*c*) Apart from the proper design of the engine as a whole, the efficiency of the unit depends to a large extent on the efficiencies of the combustion chamber and heat exchanger.

Though a combustion chamber efficiency of 0·98 has been assumed in the calculations, the actual value may be much lower, particularly for furnaces designed for low grade fuels. This will reduce the values of the efficiency indicated in the tables.

The heat exchanger efficiency is also a matter of critical interest to the hot air engine designer. The efficiency can be increased by all the usual methods such as extended surfaces, modern techniques of baffle and header design, etc. The temperature of the metal surfaces at the hot end of the heat exchanger can be kept down by a judicious combination of cross, parallel and counter flow arrangements.

Apart from these considerations, there is another factor which is present in the heat exchanger of a reciprocating hot air engine. This is the presence of pulsations in the tubes resulting from the intermittent operation of the compressor outlet valve. The effect of these pulsations can be for the better or for the worse depending upon the frequency, amplitude, flow velocity and wave velocity employed. It is shown in Part III of this Report that an increase in heat transfer coefficient of at least upto 30%, can be obtained which can be taken advantage of in the hot air engine by correspondingly reducing the heat transfer area and hence the total weight of the heat exchanger. The various findings can be incorporated into the hot air engine with the help of the design of the compressor, expansion motor and the heat exchanger headers.

The hot air engine efficiency is also increased slightly by the fact that heating takes place for part of a cycle at constant pressure, and for the remaining part at constant volume. It has been shown above that this advantage can be materialized by adjusting the phase difference between the compressor and expansion motor in such a way that in each cycle, the expansion motor inlet period precedes the compressor discharge period by a time interval determined by the length of the heat exchanger.

(*d*) While the efficiency of the power plant is low, its weight is at the same time high for a given power. Some methods of reducing the weight have been indicated in sub-section (*c*) above. Another suggestion to reduce the weight is the so-called ' two-stroke ' version of the hot air engine in which compression and expansion occur during alternative strokes of the piston in one and the same cylinder. In this version, it is possible to make provision for the larger volume required for expansion by designing the kinematic linkage of the connecting rod to give alternatively long and short strokes. While attractive from a number of points of view, it suffers from the disadvantage that the cylinder wall temperature remains essentially the same for both compression and expansion. Thus expansion occurs far below the adiabatic line, while compression occurs far above the isothermal line. The result is a considerable drop in output. At the same time, the practical

limits in the design of the kinematic linkage also limit the value of T_3 to less than 450 or 500° C. Thus at the present moment this suggestion does not seem to be practicable. The application of a stepped piston of constant stroke but with different swept volume may, on the other hand, show considerable promise.

(*e*) Assuming a simple cycle, it can be seen that the expansion and compression cylinders will have to be of different sizes in order to give maximum efficiency. Ease of manufacture can be promoted by designing the cylinders to be of the same size. It has been shown above that this can be done by separating the expansion process into two parts:—

(1) "Necessary Expansion" to produce the power required to run the compressor and overcome the losses in the unit. (2) "Useful Expansion" to produce useful power. In the range of compression ratios considered to be economical, all the three cylinders can be of the same size, if the temperature is limited to about 700° C.

(*f*) The analysis for part-load conditions shows that provision should be made for the control of the cut-off ratio and furnace temperature. The provision of speed control on the compressor introduces more complexities and also interferes with the heat transfer process in the heat exchanger.

The position may be summarized as follows:—

The efficiency of the cycle, in its most practicable form, is low and the weight of the unit is rather high. In general, the methods which improve efficiency by improving the compression and expansion process result also in increased weight of the unit as a whole. Efforts must be concentrated therefore on improving and reducing the weight of the heat exchanger and combustion chamber. In this connection, the utilisation of the pulsations present in the system is recommended both in the heat exchanger and combustion chamber.

When compared with a steam engine, the efficiency is higher but the weight is greater because of the presence of a compressor in place of a feed pump and because the air-to-air heat exchanger is bulkier than an air-to-water heat exchanger. The most important advantage of the hot air engine is that it requires no water and hence is of greater versatility and suitability in areas where water is scarce.

Compared with a closed cycle hot air engine, the cycle described here requires no cooler and is free from the danger of contamination of the working medium with lubricating oil.

Compared with internal combustion engines, the main consideration is that it is capable of accepting low-grade, cheap and locally available fuels (for which the only modifications necessary for different local conditions may concern the furnace). In fact, because of this, the efficiency or the fuel costs in relation to the output become a comparatively minor consideration.

6. Suggestions for Future Work

It is suggested that further fundamental data on the cycle may be investigated in order to make it not only practicable but to allow a design to be made which fulfils to a large extent reasonable operational expectations. This implies work along the following lines:—

(1) Building of a practicable heat exchanger making use of the pulsations present to increase the heat transfer coefficient.

(2) Studying the effect of pulsations on furnace efficiency and incorporating the results in a practical furnace.

(3) Determining the cheapest and indigenous materials that can be used in view of the fact that no objectionable gases come into contact with the stationary or especially with the moving parts of the piston-cylinder assembly.

(4) To limit the temperature of the products of combustion, and to ensure its completeness it will be necessary to admit secondary air. This can be done by an ejector working with the exhaust air of the expansion motor as pulsating forcing medium. Its design has to be investigated especially in view of the counter pressure of the combustion apparatus and the heat exchanger.

(5) Determining the best integrated design with a view to improve the ease of maintenance, operation and transportation, and to lower initial costs.

(6) Determining the cheapest and easiest starting system.

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8. List of References

1. Yellott, J. I. and Kottcamp, C. F. "Conference of Pulverised Fuel," *Inst. of Fuel*, 1947, 851.

9. List of Symbols and Units

Symbol	Significance	Units
a	Air-fuel ratio	..
B	Calorific value	kcal/kg
d	Ratio of full load to part-load	..
d_1	Ratio of full load mass flow to part-load mass flow	..
d_2	Ratio of full load temperature to part-load temperature	..
F	Amount of fuel necessary	kg/kg

Note.—Other symbols used have the same significance as in Part I.

10. List of Tables

Table XI

Values of η for two-stage compression

E	T_3 = 600	700	800	900	1000	1100	1200° K
2	4·9284	6·8862	7·6477	8·2051	8·5727	8·8507	9·0090
3	4·7311	8·2909	10·1788	11·3212	12·0588	12·5238	12·8602
4	3·5746	8·7781	10·7955	13·0846	14·0745	14·7534	15·2384
5	0·6369	7·7898	11·4122	13·5458	14·8802	15·8139	16·4565
6	..	6·9913	11·3759	13·9420	15·5883	16·6348	17·4635
7	..	5·6921	10·9433	13·9406	15·8443	17·1521	18·0885
8	..	4·2046	10·2709	13·7193	15·9392	17·4591	18·5007
9	..	2·9922	9·7696	13·5862	15·9997	17·6775	18·8898
10	..	1·3214	8·9354	13·2083	15·9045	17·7133	18·9989
11	..	..	8·2370	12·8925	15·8416	17·7912	19·2096
12	..	..	7·4122	12·4589	15·6579	17·7349	19·2199
13	..	..	6·6576	12·0924	15·4709	17·6379	19·3722
14	..	..	5·7257	11·5833	15·2507	17·6302	19·3208

TABLE XII

Values of L_s for two-stage compression

E	T_3 = 600	700	800	900	1000	1100	1200° K
2	23·0274	43·3741	63·7208	84·0675	104·4141	124·7608	145·1075
3	20·4161	50·8331	81·2501	115·7346	142·0842	172·5013	202·9183
4	14·4909	51·5746	87·0593	125·7418	162·8255	199·9091	236·9928
5	2·4368	38·2370	85·4929	127·0210	168·5490	210·0770	251·6051
6	..	30·3553	83·3206	128·4041	173·4877	218·5713	263·6548
7	..	21·6833	78·3006	126·2459	174·1912	222·1365	270·0819
8	..	15·0860	71·9917	122·3001	172·6086	222·9170	273·2255
9	..	6·5159	67·4107	119·7354	172·0601	224·3847	276·7094
10	..	..	60·5750	114·6341	168·6932	222·7522	276·8113
11	..	..	54·8797	110·4780	166·0764	221·6747	277·2731
12	..	..	48·7843	105·7594	162·7344	219·7095	276·6845
13	..	..	43·1887	101·3775	159·5670	217·7561	275·9452
14	..	..	36·6392	95·9232	155·2071	214·4911	273·7750

TABLE XIII

Values of η for two-stage compression and expansion

E	Reheat Factor	T_3 = 600	700	800	900	1000	1100	1200° K
2	1·9009	10·9002	11·3167	11·0112	10·9187	10·9027	10·9121	10·7672
	2·0	12·2675	12·4978	12·0326	11·8508	11·7776	11·7468	11·5600
3	1·8474	15·3360	15·7677	15·9183	16·0170	16·0271	15·9329	15·8544
	1·9	15·1888	16·5691	16·7884	16·8113	16·7670	16·6291	16·5177
	2·0	18·8430	18·6462	18·4425	18·3213	18·1737	17·9528	17·7786
4	1·8123	17·5036	18·3390	18·7176	18·9204	19·3029	18·8579	18·7816
	1·9	20·0864	20·4633	20·5642	20·5938	20·3768	20·3076	20·3076
	2·0	23·1602	22·8856	22·6696	22·5019	22·1314	21·9606	21·9606
5	1·7849	18·6691	20·0237	20·6227	20·9496	20·9854	21·7986	20·9745
	1·8	19·2209	20·4600	20·9981	21·2878	21·2959	21·3719	21·2504
	1·9	22·8754	23·3500	23·4847	23·5271	23·3523	23·3059	23·0777
	2·0	26·5299	26·2400	25·9713	25·7664	25·4088	25·2399	24·9049
6	1·7645	19·1160	21·0006	21·8501	22·2905	22·4688	22·4499	22·5144
	1·8	20·6028	22·1595	22·8401	23·1772	23·2830	23·2082	23·2342
	1·9	24·7907	25·4239	25·6290	25·6748	25·5765	25·3442	25·2616
	2·0	28·9786	28·6883	28·4179	28·1724	27·8700	27·4803	27·2891
7	1·7468	19·2207	21·6705	22·8015	23·3068	23·5306	23·7129	23·7491
	1·8	21·7284	23·5950	24·4351	24·7585	24·8576	24·9541	24·9220
	1·9	26·4422	27·2125	27·5060	27·4873	27·3519	27·2871	27·1265
	2·0	31·1560	30·8300	30·5768	30·2161	29·8462	29·8189	29·3311
8	1·7315	19·1306	22·0980	23·4740	24·0810	24·4639	24·7494	24·7456
	1·8	22·7264	24·8023	25·7508	26·0923	26·3029	26·4691	26·3631
	1·9	27·9756	28·7502	29·0748	29·0286	28·9875	28·9797	28·7244
	2·0	33·2249	32·6981	32·3987	31·9648	31·6721	31·4902	31·0856
9	1·7194	18·6090	19·5658	23·8794	24·6357	25·0316	25·3290	25·5570
	1·8	23·1630	25·7107	26·7229	27·1405	27·3104	27·4529	27·5678
	1·9	28·8132	29·9357	30·2508	30·2484	30·1376	30·0881	30·0627
	2·0	34·4634	34·1607	33·7788	33·3562	32·9648	32·7232	32·5576

TABLE XIII—*Contd.*

E	Reheat Factor	T_3 = 600	700	800	900	1000	1100	1200° K
10	1·7077	18·0304	22·3594	24·2108	25·1645	25·6803	25·9664	25·6308
	1·8	23·6779	26·5157	27·6598	28·1981	28·4369	28·5244	28·5090
	1·9	29·7967	31·0188	31·3965	31·4846	31·4234	31·2957	31·1162
	2·0	35·9155	35·5219	35·1331	34·7712	34·4099	34·0671	33·7233
11	1·6978	17·3346	22·3345	24·4837	25·5506	26·1957	26·5105	26·7043
	1·8	24·0268	27·1968	28·5002	29·0679	29·3906	29·4680	29·4856
	1·9	30·5749	31·9544	32·4302	32·5094	32·5167	32·3618	32·2070
	2·0	37·1231	36·7120	36·3602	35·9510	35·6429	35·2555	34·9284
12	1·6887	16·4828	24·9785	24·5822	25·7750	26·5177	26·9006	27·0507
	1·7	17·2771	22·7457	25·0454	26·1791	26·8842	27·2396	27·3682
	1·8	24·2440	27·7401	29·1446	29·7555	30·1280	30·2396	30·1781
	1·9	31·2109	32·7346	33·2437	33·3320	33·3718	33·2396	32·9880
	2·0	38·1778	37·7291	37·3428	36·9082	36·6155	36·2395	35·7979
13	1·6807	15·6094	22·0519	24·7150	26·0366	26·8218	27·2602	27·5912
	1·7	17·0331	23·0616	25·5390	26·7531	27·4695	27·8584	28·1540
	1·8	24·4092	28·2935	29·8083	30·4657	30·8253	30·9583	31·0701
	1·9	31·7852	33·5253	34·0777	34·1782	34·1812	29·9182	33·9863
	2·0	39·1613	38·7572	38·3470	37·8908	37·5370	37·1582	36·9024
14	1·6729	14·5781	21·7876	24·7358	26·2748	27·1258	27·6368	27·9310
	1·7	16·6930	23·2670	25·9364	27·3186	28·0668	28·5054	28·7450
	1·8	24·4973	28·7263	30·3667	31·1702	31·5390	31·7104	31·7489
	1·9	32·3016	34·1855	34·7971	35·0219	35·0111	34·9154	34·7527
	2·0	40·1059	39·6448	39·2275	38·8736	38·4833	38·1205	37·7565

TABLE XIV

Values of L_s for two-stage compression and expansion

E	Reheat Factor	T_3 = 600	700	800	900	1000	1100	1200° K
2	1·9009	23·4178	43·8376	64·2508	84·6641	105·0773	125·4905	145·9038
	2·0	29·8095	51·2870	72·7644	94·2419	115·7194	137·1968	158·6743
3	1·8474	21·1278	51·6705	82·2130	112·7557	142·7602	173·8409	204·3836
	1·9	26·3455	56·5872	89·1701	120·5823	151·9946	183·4069	214·8191
	2·0	36·2652	69·3307	102·3963	135·4618	168·5274	201·5438	234·6585
4	1·8123	13·0784	49·9528	86·8149	123·6771	160·5392	197·4013	234·2637
	1·9	23·7835	62·4395	101·0855	139·7315	178·3774	217·0234	255·6694
	2·0	35·9975	76·6775	117·3575	158·0375	198·7174	239·3974	280·0774
5	1·7849	2·8456	44·4544	86·0629	127·6717	169·2805	210·8893	252·4981
	1·8	4·9575	46·9183	88·8790	130·8398	172·8005	214·7613	256·7220
	1·9	18·9444	63·2363	107·5283	151·8202	196·1121	240·4040	284·6959
	2·0	32·9314	79·5544	126·1775	172·8005	219·4236	265·7497	312·6697
6	1·7645	..	37·8215	82·8633	127·5712	172·9468	217·9886	263·0304
	1·8	..	44·1650	90·1129	136·0609	182·0089	227·9569	273·9048
	1·9	13·5330	62·0336	110·5343	159·0349	207·5355	256·0362	304·5368
	2·0	28·8490	79·9023	130·9556	182·0089	233·0622	284·1155	335·1688
7	1·7468	..	30·1927	78·1334	126·0742	174·0149	221·9557	269·8964
	1·8	..	40·4132	89·8141	139·2149	188·6157	238·0165	287·4173
	1·9	7·4794	59·6247	111·7700	163·9159	216·0606	268·2059	320·3312
	2·0	23·9463	78·8361	133·7259	188·6157	243·5055	298·3953	353·2851
8	1·7315	..	22·1236	72·5086	122·8935	173·2785	223·6634	274·0488
	1·8	..	36·0765	88·4548	140·8331	193·2115	245·5898	297·9681
	1·9	1·1576	56·4458	111·7341	167·0223	222·3105	277·5988	332·8870
	2·0	18·6170	76·8152	135·0133	193·2115	251·4096	309·6077	367·8059
9	1·7194	..	14·7367	67·0345	119·3323	171·6297	223·9161	276·2254
	1·8	..	31·8977	86·6469	141·3962	196·1454	250·8946	305·6438
	1·9	..	53·1891	110·9799	168·7708	226·5616	284·3524	342·1433
	2·0	13·6480	74·4804	135·3129	196·1454	256·9778	317·8103	378·6428

TABLE XIV—*Contd.*

E	Reheat Factor	T_3 = 600	700	800	900	1000	1100	1200° K
10	1·7077	..	6·7057	60·8115	114·9179	169·0242	223·1305	278·2551
	1·8	..	27·4763	84·2070	141·2377	198·2684	255·2991	312·3298
	1·9	..	49·3549	109·5540	169·7531	229·9522	290·1512	350·3503
	2·0	8·1660	71·5335	134·9010	198·2684	261·6359	325·0033	388·3708
11	1·6978	..	..	54·8165	110·4273	166·0385	221·6493	277·2605
	1·8	..	22·6382	81·5967	140·5553	199·5138	258·4723	317·4308
	1·9	..	45·5665	107·8005	170·0345	232·2685	294·5025	356·7365
	2·0	2·9854	68·4948	134·0043	199·5138	265·0232	330·5327	396·0422
12	1·6887	..	..	48·7089	105·6964	161·2539	219·6709	276·6584
	1·7	..	4·9515	51·7597	109·1284	166·4971	223·8658	281·2345
	1·8	..	18·0134	78·7567	139·5001	200·2434	260·9867	321·7301
	1·9	..	41·6358	105·7538	169·8717	233·9897	298·1077	362·2256
	2·0	..	65·2582	132·7508	200·2434	267·736	335·2286	402·7212
13	1·6807	..	..	42·8286	100·9921	159·1556	217·3196	275·4831
	1·7	..	..	48·1719	107·0035	165·8350	224·6665	283·4981
	1·8	..	13·5651	75·8573	138·1496	200·4418	262·7340	325·0262
	1·9	..	37·7899	103·5428	169·2957	235·0486	249·9615	366·5544
	2·0	..	62·0146	131·2282	200·4418	269·6554	338·8689	408·0825
14	1·6729	..	..	36·6421	95·9435	155·2448	214·5467	273·8480
	1·7	..	..	44·3271	104·5893	164·8515	225·1138	285·3760
	1·8	..	8·8787	72·6858	136·4928	200·2999	264·1070	327·9140
	1·9	..	33·6925	101·0445	168·3964	235·7483	303·1002	370·4521
	2·0	..	58·5064	129·4032	200·2999	271·1966	342·0934	412·9901

TABLE XV

Values of η for three-stage expansion and compression

E	Reheat Factor	T_3 = 600	700	800	900	1000	1100	1200° K
2	2·8031	10·4348	10·6063	10·6076	10·6095	10·4605	10·4745	10·5177
	2·9	11·2808	11·3277	11·2512	11·2033	11·0118	11·0148	11·0275
	3·0	12·1540	12·0721	11·9154	11·8162	11·5809	11·5453	11·5536
3	2·6987	14·5440	14·9826	15·2079	15·3164	15·3337	15·3121	15·2139
	2·8	15·9820	16·1835	16·2754	16·2967	16·2505	16·1808	16·0417
	2·9	17·4013	17·3691	17·3291	17·2645	17·1556	17·0383	16·8589
	3·0	18·8206	18·5545	18·3829	18·2322	18·0605	17·8958	17·6762
4	2·6285	16·5876	17·5338	17·9511	18·0785	18·1781	18·2654	18·2672
	2·7	17·8982	18·6228	18·9121	18·9521	18·9933	19·0389	19·0064
	2·8	19·7311	20·1458	20·2557	20·1741	20·1334	20·1207	20·0404
	2·9	21·5640	21·6688	21·5996	21·3960	21·2735	21·2025	21·0743
	3·0	23·3970	23·1920	22·9433	22·6180	22·4137	22·2843	22·1083
5	2·5764	17·5860	18·9582	19·5942	19·8892	20·0655	20·2876	20·2815
	2·6	18·0949	19·3777	19·9624	20·2240	20·3771	20·5840	20·5637
	2·7	20·2518	21·1544	21·5228	21·6422	21·6974	21·8395	21·7597
	2·8	22·4086	22·9314	23·0834	23·0604	23·0177	23·0950	22·9555
	2·9	24·5650	24·7081	24·6438	24·4786	24·3379	24·3506	24·1514
	3·0	26·7219	26·4850	26·2042	25·8969	25·6582	26·6060	25·3473
6	2·5349	18·1756	19·6426	20·8445	21·2589	21·4628	21·7577	21·6926
	2·6	19·7727	20·9354	21·9852	22·2926	22·4208	22·6678	22·5548
	2·7	22·2255	22·9207	23·7377	23·8803	23·8925	24·0659	23·8792
	2·8	24·6789	24·9066	25·4902	25·4681	25·3641	25·4638	25·2036
	2·9	27·1322	26·8919	27·2428	27·0559	26·8356	26·8617	26·5280
	3·0	29·5855	28·8776	28·9953	28·6439	28·3071	28·2598	27·8524
7	2·5012	18·2882	20·4955	21·6241	22·1391	22·5098	22·6810	22·7992
	2·6	20·9445	22·6490	23·5055	23·8382	24·0886	24·1671	24·2153
	2·7	23·6330	24·8286	25·4100	25·5583	25·6866	25·6712	25·6485
	2·8	26·3214	27·0081	27·3145	27·2781	27·2845	27·1754	27·0818
	2·9	29·0104	29·1876	29·2191	28·9889	28·8827	30·1836	28·5151
	3·0	31·6989	31·3672	31·1236	30·7180	30·4807	30·1836	29·9483

E	Reheat Factor	T_3 = 600	700	800	900	1000	1100	1200° K
8	2·4719	18·2898	20·9331	22·2525	23·0439	23·3796	23·5593	23·8525
	2·5	19·1112	21·5947	22·8281	23·5656	23·8614	24·0113	24·2852
	2·6	22·0356	23·9487	24·8749	25·4226	25·5753	25·6196	25·8254
	2·7	24·9600	26·3031	20·2822	27·2796	27·2895	27·2281	27·3658
	2·8	27·8682	28·6571	28·7929	29·1368	29·0034	28·8365	28·9059
	2·9	30·8088	31·0115	31·0165	30·9938	30·7176	30·4450	30·4461
	3·0	33·7338	33·3655	33·0633	32·5084	32·4315	32·0535	31·9863
9	2·4467	18·1439	21·3544	22·7483	23·5938	24·1483	24·4475	24·6161
	2·5	19·8110	22·6012	23·9042	24·6372	25·1157	25·3570	25·4801
	2·6	22·9511	25·1180	26·0806	26·6019	26·9376	27·0698	27·1078
	2·7	26·0904	27·6349	28·2573	28·5665	28·7593	28·7824	28·7354
	2·8	29·2298	30·1517	30·4337	30·5312	30·5812	30·4951	30·3630
	2·9	32·3692	32·6686	32·6104	32·4958	32·4031	32·2077	31·9907
	3·0	35·5092	35·1855	34·7868	34·4605	34·2248	33·9205	33·6183
10	2·4252	17·8783	21·3887	23·0771	24·0094	24·7336	24·9879	25·1256
	2·5	20·3703	23·3725	24·7882	25·5493	26·1654	26·3275	26·3946
	2·6	23·7017	26·0246	27·0756	26·6083	28·0796	28·1186	28·0912
	2·7	27·0331	28·6768	29·3634	27·6669	29·9939	29·9099	29·7878
	2·8	30·3639	31·3289	31·6508	31·7258	31·9082	31·7010	31·4843
	2·9	33·6953	33·9810	33·9383	33·7844	33·8224	33·4921	33·1810
	3·0	37·0267	36·6331	36·2257	35·8433	35·7364	35·2832	34·8775
11	2·4061	17·5199	21·4676	23·3472	24·4325	25·1229	25·5377	25·7861
	2·5	20·8166	24·0819	25·5953	26·4559	26·9941	27·2948	27·4546
	2·6	24·3243	26·8630	27·9874	28·6085	28·9844	29·1643	29·2295
	2·7	27·8320	29·6445	30·3791	30·7612	30·9750	31·0336	31·0044
	2·8	31·3390	32·4255	32·7708	32·9138	32·9656	32·9031	32·7795
	2·9	34·8467	35·2071	35·1625	35·0664	34·9563	34·7723	34·5544
	3·0	38·3544	37·9881	37·5546	37·2190	36·9469	36·6418	36·3295

E	Reheat Factor	T_3 = 600	700	800	900	1000	1100	1200° K
12	2·3883	17·0949	21·4556	23·5554	24·6975	25·4219	25·9728	26·3101
	2·5	21·1989	24·6942	26·3364	27·1912	27·7221	28·1390	28·3698
	2·6	24·8727	27·5939	28·8266	29·4236	29·7813	30·0781	30·2139
	2·7	28·5465	30·4932	31·3163	31·6563	31·8405	32·0172	32·0578
	2·8	32·2203	33·3925	33·8061	33·8886	33·9000	33·9565	33·9017
	2·9	35·8941	36·2917	36·2958	36·1210	35·9592	35·8956	35·7458
	3·0	39·5687	39·1915	38·7860	38·3533	38·0184	37·8347	37·5898
13	2·3728	16·6180	21·3220	23·6800	24·9203	25·8482	26·3245	26·5983
	2·5	21·4795	25·1333	26·9564	27·8533	28·5654	28·8693	29·0084
	2·6	25·3015	28·1295	29·5323	30·1591	30·7017	30·8701	30·9031
	2·7	29·1235	31·1256	32·1081	32·4645	32·8379	32·8709	32·7979
	2·8	32·9455	34·1218	34·6840	34·7702	34·9742	34·8717	34·6926
	2·9	36·7675	37·1179	37·2599	37·0760	37·1104	36·8724	36·5874
	3·0	40·5895	40·1141	39·8357	39·3817	39·2466	38·8732	38·4822
14	2·3580	16·1070	21·2554	23·7103	25·1833	26·0563	26·6489	26·8660
	2·4	17·7769	22·5605	24·8249	26·1852	26·9790	27·5147	27·6828
	2·5	21·7536	25·6684	27·4795	28·5698	29·1754	29·5762	29·6276
	2·6	25·7312	28·7757	30·1337	30·9544	31·3722	31·6375	31·5721
	2·7	29·7088	31·8831	32·7884	33·3394	33·5687	33·6991	33·5169
	2·8	33·6846	34·9910	35·4426	35·7241	35·7654	35·7604	35·4617
	2·9	37·6622	38·0983	38·0968	38·1090	37·9619	37·8220	37·4063
	3·0	41·6389	41·2056	40·7514	40·4937	40·1587	39·8833	39·3511

TABLE XVI

Values of L_t for three-stage compression and expansion

E	Reheat Factor	T_3 = 600	700	800	900	1000	1100	1200° K
2	2·8031	25·1862	45·5816	65·9779	86·3733	106·7687	127·1650	147·5604
	2·9	29·4161	50·5175	71·6180	92·7186	113·8192	134·9205	156·0211
	3·0	33·7818	55·6109	77·4392	99·2675	121·0958	142·9241	164·7524
3	2·6987	25·7572	56·1540	86·5507	116·9475	147·3443	177·7411	208·1378
	2·8	32·6041	64·1409	95·6791	127·2159	158·7541	190·2922	221·8291
	2·9	39·3617	72·0261	104·6892	137·3536	170·0683	202·6812	235·3460
	3·0	46·1193	79·9100	113·7007	147·4901	181·2808	215·0714	248·7829
4	2·6285	20·4427	57·3890	94·3338	131·2801	168·2249	205·1712	242·1160
	2·7	26·4731	64·4240	102·3749	140·3241	178·2750	216·2259	254·1752
	2·8	34·9066	74·2625	113·6185	152·9744	192·3304	231·6863	271·0422
	2·9	43·3401	84·1011	124·8637	165·6247	206·3857	247·1483	287·9093
	3·0	51·7737	93·9413	136·1074	178·2750	220·4427	262·6087	304·7763
5	2·5764	13·3397	54·7279	96·1181	137·5064	178·8965	220·2848	256·8150
	2·6	15·6142	57·3834	99·1508	140·9200	182·6874	224·4566	266·2240
	2·7	25·2535	68·8164	112·0020	155·3781	198·7523	242·1266	285·5027
	2·8	34·8929	79·8740	124·8551	169·8362	214·8173	259·7984	304·7795
	2·9	44·5303	91·1183	137·7063	184·2943	230·8822	277·4702	324·0563
	3·0	54·1697	102·3645	150·5575	198·7523	246·9472	295·1401	343·3350
6	2·5349	5·4591	48·8814	95·5379	140·5751	185·6145	230·6539	278·6933
	2·6	12·3994	56·9373	104·7901	150·9855	197·1809	243·3763	289·5738
	2·7	23·0584	69·3090	114·0043	166·9761	214·9501	262·9220	310·8939
	2·8	33·7195	81·6850	133·2184	182·9668	232·7173	282·4656	332·2140
	2·9	44·3806	94·0568	147·4325	198·9574	250·4844	302·0093	353·5363
	3·0	55·0418	106·4306	161·6466	214·9501	268·2515	321·5550	374·8464
7	2·5012	..	45·5207	93·3197	141·1186	188·9176	236·7166	284·5155
	2·6	9·0503	58·7378	108·4229	158·1103	207·7978	257·4852	307·1726
	2·7	20·5158	72·1149	123·7117	175·3108	226·9076	278·5067	330·1035
	2·8	31·9812	85·4920	139·0005	192·5090	246·0174	299·5282	353·0367
	2·9	43·4490	98·8691	154·2893	209·6166	265·1296	320·5498	375·9699
	3·0	54·9144	112·2462	169·5781	226·9076	284·2394	341·5713	398·9008

TABLE XVI—*Contd.*

E	Reheat Factor	T_3 = 600	700	800	900	1000	1100	1200° K
8	2·4719	..	39·9714	90·2517	140·5345	190·8148	241·0976	291·3780
	2·5	..	43·9727	94·8265	145·6777	196·5314	247·3852	298·2364
	2·6	5·6390	58·2105	111·0975	163·9845	216·8714	269·7584	322·6454
	2·7	17·5280	72·4507	124·5892	182·2912	237·2139	294·3878	347·0569
	2·8	29·7325	86·6885	142·2422	200·6004	257·5539	314·5099	371·4659
	2·9	41·9370	100·9287	159·9180	218·9072	277·8964	336·8857	395·8749
	3·0	54·1440	115·1665	176·1890	237·2139	298·2364	359·2614	420·2839
9	2·4469	..	34·8470	86·6882	139·0627	191·4373	243·8118	296·1890
	2·5	..	42·2693	95·7826	149·2933	202·8040	256·3147	309·8254
	2·6	1·6029	57·2525	112·9048	168·5571	224·2094	280·1283	335·5113
	2·7	14·4445	72·2357	130·0296	187·8208	245·6120	303·4059	361·1971
	2·8	27·2861	87·2189	147·1517	207·0845	267·0174	326·9502	386·8830
	2·9	40·1277	102·2021	164·2765	226·3483	288·4227	350·4945	412·5689
	3·0	52·9720	117·1853	181·3987	245·6120	309·8254	374·0414	437·4597
10	2·4252	..	28·7539	82·8444	136·9349	191·0254	245·1159	299·2064
	2·5	..	40·4324	96·1908	151·9493	207·7078	263·4662	319·2247
	2·6	..	56·0449	114·0329	172·0237	230·0117	287·9997	345·9905
	2·7	11·4398	71·6574	131·8777	192·0953	252·3156	312·5360	372·7536
	2·8	24·8199	87·2698	149·7198	212·1697	274·6196	337·0695	399·5166
	2·9	38·2028	102·8823	167·5618	232·2413	296·9235	361·6030	426·2825
	3·0	51·5858	118·4948	185·4038	252·3156	319·2247	386·1365	453·0455
11	2·4060	..	23·2396	78·8559	134·4723	190·0886	245·7049	301·3213
	2·5	..	38·4496	96·2371	154·0276	211·8180	269·6056	327·3960
	2·6	..	54·6292	114·7318	174·8316	234·9313	295·0340	355·1337
	2·7	8·3997	70·8117	133·2236	195·6355	258·0475	320·4594	383·8713
	2·8	22·2671	86·9912	151·7154	216·4395	281·1637	345·8878	410·6119
	2·9	36·1374	103·1737	170·2072	237·2435	304·2798	371·3133	438·3496
	3·0	50·0077	119·3533	188·7019	258·0475	327·3960	396·7416	466·0902

TABLE XVI—*Contd.*

E	Reheat Factor	T_s = 600	700	800	900	1000	1100	1200° K
12	2·3883	..	17·7489	74·7009	131·6497	188·4956	245·5505	302·5024
	2·5	..	36·3936	96·0081	155·6227	215·2373	274·8518	334·4633
	2·6	..	53·0879	115·0869	177·0828	239·0819	301·0809	363·0799
	2·7	5·3956	69·7791	134·1626	198·5460	262·9265	328·3909	391·6934
	2·8	19·7034	86·4703	153·2382	220·0062	286·7741	353·5421	420·3069
	2·9	34·0091	103·1615	172·3139	241·4663	310·6187	379·7711	448·9235
	3·0	48·3189	119·8558	191·3927	262·9265	334·4633	406·0002	477·5370
13	2·3728	..	12·5239	70·6523	128·7775	186·9027	245·0311	303·1564
	2·5	..	34·3371	95·5798	156·8226	218·0622	279·3049	340·5477
	2·6	..	51·4851	115·1775	178·8700	242·5593	306·2517	369·9442
	2·7	2·4909	68·6330	134·7752	200·9142	267·0564	333·1985	399·3407
	2·8	17·1891	85·7810	154·3729	222·9616	291·5535	439·2704	428·7372
	2·9	31·8874	102·9290	173·9706	245·0090	316·0506	387·0922	458·1338
	3·0	46·5856	120·0769	196·7332	267·0564	340·5477	414·0390	487·5303
14	2·3580	..	7·1177	66·4131	125·7052	185·0006	244·2961	303·5882
	2·4	..	14·5104	74·8600	135·2129	195·5626	255·9122	316·2619
	2·5	..	32·1147	94·9787	157·8428	220·7069	283·5742	346·4383
	2·6	..	49·7157	115·0942	180·4727	245·8545	311·2330	376·6115
	2·7	..	67·3167	135·2129	203·1059	270·9988	338·8950	406·7880
	2·8	14·5104	84·9210	155·3284	225·7358	296·1464	366·5538	436·9644
	2·9	29·6002	102·5220	175·4438	248·3689	321·2907	394·2158	467·1376
	3·0	44·6868	120·1231	195·5626	270·9988	346·4383	421·8745	497·3141

TABLE XVII

Amount of air necessary for combustion

a	E	T_s = 600	800	1000	1200° K
1·0	2	0·1192	0·2271	0·3394	0·4556
	3	0·0948	0·2027	0·3150	0·4313
	4	0·0754	0·1831	0·2953	0·4114
	5	0·0592	0·1668	0·2789	0·3950
	6	0·0452	0·1531	0·2655	0·3817
	7	0·0326	0·1405	0·2529	0·3691
	8	0·0211	0·1287	0·2409	0·3569
	9	0·0106	0·1183	0·2305	0·3466
	10	..	0·1086	0·2209	0·3370
	11	..	0·0996	0·2120	0·3280
	12	..	0·0906	0·2027	0·3188
	13	..	0·0761	0·1948	0·3110
	14	..	0·0747	0·1868	0·3029
1·2	2	0·1419	0·2703	0·4040	0·5424
	3	0·1129	0·2413	0·3750	0·5134
	4	0·0897	0·2180	0·3516	0·4898
	5	0·0704	0·1986	0·3321	0·4702
	6	0·0538	0·1823	0·3161	0·4545
	7	0·388	0·1672	0·3010	0·4395
	8	0·0251	0·1533	0·2868	0·4249
	9	0·0126	0·1408	0·2744	0·4126
	10		0·1293	0·2629	0·4012
	11	..	0·1186	0·2524	0·3905
	12	..	0·1078	0·2414	0·3796
	13	..	0·0906	0·2319	0·3702
	14	..	0·0890	0·2224	0·3606
1·4	2	0·1703	0·3244	0·4848	0·6508
	3	0·1354	0·2895	0·4500	0·6161
	4	0·1077	0·2616	0·4219	0·5877
	5	0·0845	0·2383	0·3985	0·5642
	6	0·0645	0·2187	0·3793	0·5453
	7	0·0465	0·2007	0·3612	0·5273
1·4	8	0·0301	0·1839	0·3441	0·5099
	9	0·0151	0·1690	0·3293	0·4951
	10	..	0·1551	0·3155	0·4815
	11	..	0·1424	0·3029	0·4686
	12	..	0·1294	0·2896	0·4555
	13	..	0·1088	0·2783	0·4442
	14	..	0·1067	0·2669	0·4327

TABLE XVII—*Contd.*

a	E	$T_3 = 600$	800	1000	1200° K
1·6	2	0·1930	0·3676	0·5495	0·7376
	3	0·1535	0·3281	0·5100	0·6982
	4	0·1220	0·2964	0·4781	0·6661
	5	0·0958	0·2701	0·4516	0·6395
	6	0·0731	0·2479	0·4298	0·6181
	7	0·0527	0·2274	0·4094	0·5977
	8	0·0341	0·2084	0·3900	0·5779
	9	0·0171	0·1915	0·3732	0·5611
	10	..	0·1758	0·3576	0·5457
	11	..	0·1613	0·3432	0·5311
	12	..	0·1466	0·3282	0·5162
	13	..	0·1233	0·3154	0·5035
	14	..	0·1210	0·3025	0·4903
1·8	2	0·2157	0·4109	0·6141	0·8244
	3	0·1715	0·3667	0·5700	0·7804
	4	0·1079	0·3313	0·5244	0·7445
	5	0·1070	0·3019	0·5048	0·7147
	6	0·0817	0·2770	0·4804	0·6908
	7	0·0589	0·2542	0·4576	0·6680
	8	0·0382	0·2329	0·4359	0·6459
	9	0·0191	0·2141	0·4171	0·6272
	10	..	0·1965	0·3996	0·6099
	11	..	0·1803	0·3836	0·5936
	12	..	0·1639	0·3669	0·5769
	13	..	0·1378	0·3525	0·5627
1·8	14	..	0·1352	0·3381	0·5480
2·0	2	0·2384	0·4541	0·6788	0·9111
	3	0·1896	0·4053	0·6300	0·8625
	4	0·1507	0·3662	0·5906	0·8228
	5	0·1183	0·3336	0·5579	0·7900
	6	0·0903	0·3062	0·5310	0·7635
	7	0·0651	0·2809	0·5057	0·7383
	8	0·0422	0·2575	0·4818	0·7139
	9	0·0211	0·2366	0·4610	0·6932
	10	..	0·2172	0·4417	0·6741
	11	..	0·1993	0·4240	0·6560
	12	..	0·1811	0·4055	0·6376
	13	..	0·1523	0·3896	0·6219
	14	..	0·1494	0·3737	0·6057

TABLE XVII—*Contd.*

a	E	$T_3 = 600$	800	1000	1200° K
2·1	2	0·2611	0·4974	0·7434	0·9979
	3	0·2076	0·4439	0·6900	0·9447
	4	0·1651	0·4011	0·6469	0·9012
	5	0·1296	0·3654	0·6110	0·8652
	6	0·0989	0·3353	0·5815	0·8362
	7	0·0713	0·3077	0·5539	0·8086
	8	0·0462	0·2820	0·5277	0·7819
	9	0·0231	0·2591	0·5049	0·7592
	10	..	0·2379	0·4838	0·7383
	11	..	0·2183	0·4644	0·7185
	12	..	0·1984	0·4441	0·6984
	13	..	0·1668	0·4267	0·6812
	14	..	0·1637	0·4093	0·6634
2·4	2	0·2894	0·5514	0·8242	1·1064
	3	0·2302	0·4922	0·7651	1·0473
	4	0·1830	0·4447	0·7172	0·9991
	5	0·1437	0·4051	0·6774	0·9592
	6	0·1097	0·3718	0·6447	0·9271
2·4	7	0·0791	0·3411	0·6141	0·8965
	8	0·0512	0·3126	0·5850	0·8668
	9	0·0257	0·2873	0·5598	0·8417
	10	..	0·2637	0·5364	0·8185
	11	..	0·2420	0·5148	0·7966
	12	..	0·2200	0·4924	0·7743
	13	..	0·1849	0·4731	0·7552
	14	..	0·1815	0·4537	0·7355
2·6	2	0·3121	0·5947	0·8889	1·1932
	3	0·2483	0·5308	0·8251	1·1295
	4	0·1974	0·4795	0·7734	1·0776
	5	0·1549	0·4369	0·7306	1·0344
	6	0·1183	0·4010	0·6953	0·9998
	7	0·0853	0·3679	0·6623	0·9668
	8	0·0552	0·3372	0·6309	0·9348
	9	0·277	0·3098	0·6037	0·9077
	10	..	0·2844	0·5784	0·8827
	11	..	0·2610	0·5552	0·8591
	12	..	0·2372	0·5310	0·8350
	13	..	0·1994	0·5102	0·8144
	14	..	0·1957	0·4893	0·7932

Table XVIII

Thermal efficiency at part-load

d_1	T_3, ° K	E	$d_2 = 1{\cdot}000$	1·333	1·500	2·000
1·000	600	2	0·0414	..	..	..
		3	0·0154	..	..	..
	800	2	0·0751	0·0414	0·0115	..
		3	0·0969	0·0155	..	..
		4	0·0981	..	..	..
		5	0·0895	..	..	..
		6	0·0725	..	..	..
		7	0·0489	..	..	..
		8	0·0195	..	..	..
	1000	2	0·0872	0·0700	0·0577	..
		3	0·1221	0·0859	0·0573	..
		4	0·1381	0·0795	0·0282	..
		5	0·1455	0·0621	..	..
		6	0·1470	0·0339	..	..
		7	0·1447	..	..	..
		8	0·1398	..	..	..
		9	0·1326	..	..	..
		10	0·1237	..	..	..
		11	0·1131	..	..	..
		12	0·1009	..	..	..
		13	0·0876	..	..	..
		14	0·0730	..	..	..
	1200	2	0·0935	..	..	..
		3	0·1343	0·1122	0·0969	0·0154
		4	0·1565	0·1227	0·0981	..
		5	0·1698	0·1244	0·0895	..
		6	0·1778	0·1196	0·0725	..
		7	0·1822	0·1104	0·0489	..
		8	0·1843	0·0978	0·0195	..
		9	0·1845	0·0821	..	..
		10	0·1835	0·0636	..	..
		11	0·1813	0·0697	..	..
1·000	1200	12	0·1783	0·0181	..	..
		13	0·1745	..	..	..
		14	0·1701	..	..	..

Table XVII—*Contd.*

a	E	$T_3 = 600$	800	1000	1200° K
2·8	2	0·3348	0·6379	0·9535	1·2799
	3	0·2663	0·5694	0·8851	1·2116
	4	0·2118	0·5144	0·8297	1·1559
	5	0·1662	0·4687	0·7837	1·1097
	6	0·1269	0·4301	0·7459	1·0725
	7	0·0915	0·3947	0·7104	1·0371
	8	0·0592	0·3617	0·6768	1·0028
	9	0·0297	0·3323	0·6476	0·9737
	10	..	0·3051	0·6205	0·9469
	11	..	0·2800	0·5956	0·9216
	12	..	0·2545	0·5696	0·8957
	13	..	0·2139	0·5473	0·8737
	14	..	0·2099	0·5249	0·8509

TABLE XVIII—*Contd.*

d_1	T_3, °K	E	$d_2 = 1{\cdot}000$	1·333	1·500	2·000
1·333	600	2	0·0311	..	..	..
		3	0·0116	..	..	..
	800	2	0·0563	0·0311	0·0086	..
		3	0·0727	0·0116	..	..
		4	0·0736	..	..	..
		5	0·0671	..	..	..
		6	0·0544	..	..	..
		7	0·0367	..	..	..
		8	0·0146	..	..	..
	1000	2	0·0654	0·0525	0·0433	..
		3	0·0916	0·0644	0·0430	..
		4	0·1036	0·0596	0·0212	..
		5	0·1092	0·0466	..	..
		6	0·1103	0·0254	..	..
		7	0·1086	..	..	..
		8	0·1049	..	..	..
		9	0·0995	..	..	..
		10	0·0928	..	..	..
		11	0·0848	..	..	..
		12	0·0757	..	..	..
		13	0·0657	..	..	..
		14	0·0548	..	..	..
	1200	2	0·0701	0·0617	0·0563	0·0311
		3	0·1008	0·0842	0·0727	0·0116
		4	0·1174	0·0920	0·0736	..
		5	0·1274	0·0933	0·0671	..
		6	0·1334	0·0897	0·0544	..
		7	0·1367	0·0828	0·0367	..
		8	0·1383	0·0734	0·0146	..
		9	0·1384	0·0616	..	..
1·333	1200	10	0·1377	0·0477	..	..
		11	0·1360	0·0523	..	..
		12	0·1338	0·0136	..	..
		13	0·1309	..	..	..
		14	0·1276	..	..	..
2·000	600	2	0·0207	..	..	..
		3	0·0077	..	..	..

d_1	T_3, °K	E	$d_2 = 1{\cdot}000$	1·333	1·500	2·000
	800	2	0·0376	0·0207	0·0058	..
		3	0·0485	0·0078	..	..
		4	0·0491	..	..	..
		5	0·0448	..	..	..
		6	0·0363	..	..	..
		7	0·0245	..	..	..
		8	0·0098	..	..	..
	1000	2	0·0436	0·0350	0·0289	..
		3	0·0611	0·0430	0·0287	..
		4	0·0691	0·0398	0·0141	..
		5	0·0728	0·0311	..	..
		6	0·0735	0·0170	..	..
		7	0·0724	..	..	..
		8	0·0699	..	..	..
		9	0·0663	0·0350	0·0289	..
		10	0·0619	0·0430	0·0287	..
		11	0·0566	0·0398	0·0141	..
		12	0·0505	0·0311	..	..
		13	0·0438	0·0170	..	..
		14	0·0365	..	..	..
	1200	2	0·0468	0·0412	0·0375	0·0207
		3	0·0672	0·0561	0·0485	0·0077
		4	0·0783	0·0614	0·0491	..
		5	0·0849	0·0622	0·0448	..
		6	0·0889	0·0598	0·0363	..
2·000	1200	7	0·0911	0·0552	0·0245	..
		8	0·0922	0·0489	0·0098	..
		9	0·0923	0·0411	..	..
		10	0·0918	0·0318	..	..
		11	0·0907	0·0349	..	..
		12	0·0892	0·0091	..	..
		13	0·0873	..	..	..
		14	0·0851	..	..	..
4·000	600	2	0·0104	..	..	..
		3	0·0039	..	..	..

TABLE XVIII—*Contd.*

d_1	T_3, °K.	E	$d_2 = 1\cdot000$	1·333	1·500	2·000
	800	2	0·0188	0·0104	0·0029	..
		3	0·0242	0·0039	..	..
		4	0·0245	..	..	..
		5	0·0224	..	..	..
		6	0·0181	..	..	..
		7	0·0122	..	..	..
		8	0·0049	..	..	..
	1000	2	0·0218	0·0175	0·0144	..
		3	0·0305	0·0215	0·0143	..
		4	0·0345	0·0199	0–0071	..
		5	0·0364	0·0155	..	..
		6	0·0368	0·0085	..	..
		7	0·0362	..	..	..
		8	0·0350	..	..	..
		9	0·0332	..	..	..
		10	0·0309	..	..	..
		11	0·0283	..	..	..
		12	0·0252	..	..	..
		13	0·0219	..	..	..
		14	0·0183	..	..	..
4·000	1200	2	0·0234	0·0206	0·0188	0·0104
		3	0·0336	0·0281	0·0242	0·0039
		4	0·0391	0·0307	0·0245	..
		5	0·0425	0·0311	0·0244	..
		6	0·0445	0·0299	0·0181	..
		7	0·0456	0·0276	0·0122	..
		8	0·0461	0·0245	0·0049	..
		9	0·0461	0·0205	..	..
		10	0·0459	0·0159	..	..
		11	0·0453	0·0174	..	..
		12	0·0446	0·0045	..	..
		13	0·0436	..	..	..
		14	0·0425	..	..	..

STUDIES FOR A NEW HOT AIR ENGINE

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Foreword

The investigation on the rate of heat transfer to be expected in the new hot air engine is the second account of work initiated and undertaken in the Internal Combustion Engineering Department under the auspices of the Council of Scientific and Industrial Research within the research scheme on "Hot Air Engines and Their Development". As were the first and second parts, it is devoted to fundamental data and thus forms the third part of the studies for a New Hot Air Engine which are continued and will be reported in this Journal in due course.

PART III:

Fundamentals of Heat Exchanger Operation with Oscillating Flow

Summary

The problem involved in the design and operation of the heat exchanger of the New Hot Air Engine is the determination of the rate of heat transfer between a hot tube and air flowing in it turbulently, when pulsations are imposed on the air.

A model heat transfer investigation was conducted with air flowing unsteadily in, and completely interrupted by, pulses through a horizontal brass pipe of 1″ inside diameter which was heated externally by steam. The Reynolds numbers calculated for equivalent steady flow conditions ranged from 6,000–25,000 and the frequencies from 5–40 c/s. An increase in the rate of heat transfer was observed only within a limited range of frequencies and at certain Reynolds numbers. A marked improvement of the rate of heat transferred could be obtained by fitting an orifice plate as a constriction to the end of the heat transfer pipe whereby pressure waves were reflected and the previously obtained dependence of air pressure on time and locality was altered.

CONTENTS

1. Introduction

A study of the hot air engine cycle described in the previous parts of this report shows that the efficiency of the heat exchanger is one of the most important factors deciding the overall efficiency of the power plant. An increase of the heat exchanger efficiency from 80 to 85% is equivalent to raising the maximum temperature of the cycle by nearly 150° C. The heat exchanger also adds bulk to the hot air engine and any increase in efficiency can be directly utilised for its reduction.

The heat exchanger in a hot air engine enjoys a particular mode of operation not usual for normal heat exchangers. This is the occurrence particularly on the cold side, of pressure pulsations of the air flow. It therefore becomes desirable to investigate the effect of these pulsations on the heat transfer coefficient. The report presented here examines whether pulsations imposed on, and completely interrupting, a flowing fluid with varying pressure amplitude and frequency have any effect on the rate of heat transfer to this fluid.

2. Previous Work

Periodical flow phenomena and their theoretical treatment have been outlined by Schlichting[1] with particular reference to boundary layers. The flow pattern near an oscillating wall has also been given by the same author. The peculiar distribution of the velocity across the radius of a pipe (Fig. 1) has been underlined by Richardson,[2] theoretically derived by the same author[3] and by Th. Sexl,[4] and confirmed by E. Taylor.[5] The maximum occurs near the wall and not in the centre of the pipe, and the slope of the profile within the boundary layer is different from that in Poiseuille's flow.

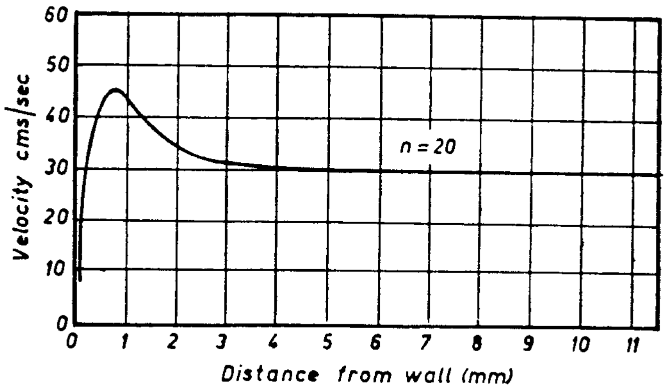


Fig. 1. A Typical Velocity Profile in Laminar Pulsating Flow.

Periodical or instationary heat transfer phenomena have first been approached by R. Sinn[6] in experiments undertaken on a gas column made to oscillate by a moving piston, and theoretically by Pfriem.[7] The heat transfer from air oscillating with high amplitudes in a closed vessel was determined by Havemann[8] and a considerable increase of the rate of heat transfer was found. P. L. Kapitza[9] has established relations describing heat conduction and diffusion in liquids with periodic flow as also for wave flow of viscous liquids. K. Elser[10] has calculated the heat transfer of a periodically adiabatically compressed turbulent gas to the wall whereby, however, the gas pressure depends solely on time.

From a consideration of proportionality between the velocity and temperature fields near a heat exchanging surface, it may be deduced that the rate of heat transfer in the case of pulsating flow would be different from that encountered for steady flow. It is also possible that the thickness of the boundary layer for pulsating flow is subject to variations due to varying pressures acting on its surface. Thus the thermal resistance would change accordingly and the heat transfer process may be affected. Furthermore, the mean boundary layer thickness may depend on the rate of change of pressure in the wave front rather than on the amplitude alone. Hence the shape of the pressure wave could be expected to modify the rate of heat transfer. Finally, turbulent exchange processes may decisively be affected by the local changes of pressure with time as also by pressure differences along the wall of the pipe.

Reference to previous literature shows that experimental evidence for the change of the rate of heat transfer is meagre and sometimes conflicting. The fluids used have been water, oil and air.

2.1. *Water.*

Turbulent Flow.—Martinelli *et al.* [11, 12] studied the effect of pulsations with frequencies of 0·22 to 4·4 c/s. when water was heated in a vertical tube between Reynolds numbers of 2,660 and 77,300. Marchant[13] worked with frequencies of 0·167, 0·417 and 1·0 c/s. and Reynolds numbers of 200 to 100,000. Both reported no difference in heat transfer coefficients between steady and pulsating flow but West and Taylor[14] reported an increase of 60 to 70% in the range $30{,}000 < Re < 85{,}000$ at a frequency of 1·67 c/s. The difference of the two results may be due to the fact that Martinelli and Marchant had made use of the pulsations produced by a reciprocating compressor while West and Taylor had introduced between the compressor and the heat transfer section, a receiver chamber where the pressure could be varied at will. Thus the shape of the velocity wave was modified and the increase in rate of heat transfer was obtained only at certain pressures in the receiver chamber. Andreas[15] advocated a mechanical vibration of the entire heat exchanger with a minimum frequency of 25 c/s. and amplitudes of 1 to 5 mm. in the direction of the tubes.

Laminar Flow.—Marchant[13] noted an increase in heat transfer in the range of Reynolds numbers from 400 to 2,000 at frequencies of 0·167, 0·417 and 1·0 c/s. Morris[16] and Webb[17] obtained no change of the rate of heat transfer in the range $26 < Re < 1{,}375$. These differing results might be due to a difference in the shape of the pressure waves.

2.2. *Oil.*

Turbulent Flow.—L i n k e[18] investigated the effect of pulsations on the performance of oil coolers and found an increase in heat transfer of up to 35%.

Laminar Flow.—L i n k e[18] also found an increase of up to 300% in the laminar region. This great increase may be expected from theoretical considerations although it is contradicted by the data for water given above. The imposition of pulsations on laminar flow is in effect, equivalent to the initiation of some type of turbulence in the flow as has been shown by R i c h a r d s o n.[2] The rate of heat transfer may therefore be expected to increase considerably provided other conditions, namely the shape, frequency and amplitude of the pressure wave, are suitably chosen.

2.3. *Air.*

Turbulent Flow.—S t a n t o n *et al.*[19] investigated the effect of cylinder vibrations with a frequency of 50 c/s. on thermal emissivity of fin surfaces on an aircraft engine cylinder and found that they had no influence on heat transfer. This was probably due to the fact that the amplitude of the vibrations was very small.

Laminar Flow.—K u b a n s k i i[20, 21] imposed acoustic vibrations of frequencies ranging from 8 to 30 kc/s. on air flowing in a pipe at Reynolds numbers below 2,500 and obtained an increase in heat transfer of up to 50%.

The results of previous work together with experimental data are summarised in Table I which shows that most of the work reported in this field so far has been concerned with incompressible fluids like oil or water.

Compressibility does not appear to play any appreciable part for laminar flow in any fluid at the velocities which have been investigated. In the case of pulsating flow, however, compressibility may be expected to affect the heat transfer process as every pressure wave will compress the boundary layer and besides, will be immediately followed by a rarefaction wave. As this effect is absent in incompressible fluids, the results obtained with water and air, for instance, may not be comparable. It is further to be noted that no author has specifically reported an actual reduction of the heat transfer coefficient; this possibility exists if compressibility were to be taken into account.

3. Scope of Work

The work presented here seeks to clarify the rate of heat transfer from a horizontal tube heated externally by condensing steam, to turbulent air flowing inside at almost atmospheric pressure and with pulsations of varying frequencies and amplitudes imposed on it.

Flow pulsations were produced by a poppet valve in preference to a rotary or piston valve in order to simulate the type of pressure wave to be expected in the hot air engine. In the experiments recorded here the Reynolds numbers varied in the range from 6,000 to 25,000 and the frequencies of pulsations in the range 5 to 40 c/s.

Table I

Tabular Representation of the Results of Previous Work

Sl. No.	Fluid	Flow condition	Reynolds number	Frequency of pulsations c/s.	Amplitude cm.	Difference in heat transfer	Reference
1	Water	Turbulent	2660–77300	0·22–4·4		Negligible	12
2	Water	Turbulent	2000–100,000	0·167; 0·417; 1·0		Negligible	13
3	Water	Turbulent	30000–85000	1·67		+60 to 70%	14
4	Water	Turbulent		25	0·1–0·5	Increase	15
5	Water	Laminar	400–2000	0·167; 0·417; 1·0		Increase	13
6	Water	Laminar	26–1375			Negligible	16, 17
7	Oil	Turbulent				+ 35%	18
8	Oil	Laminar				+ 300%	18
9	Air	Turbulent		50	Very small	Negligible	19
10	Air	Laminar	2500	8–30 kc/s.		+ 50%	21

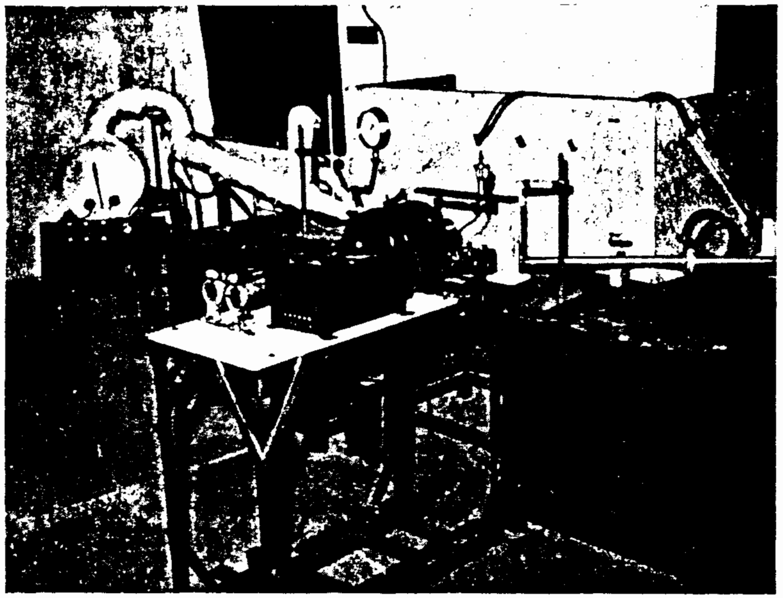


FIG. 2. A View of the Experimental Set-up for the Measurement of Heat Transfer.

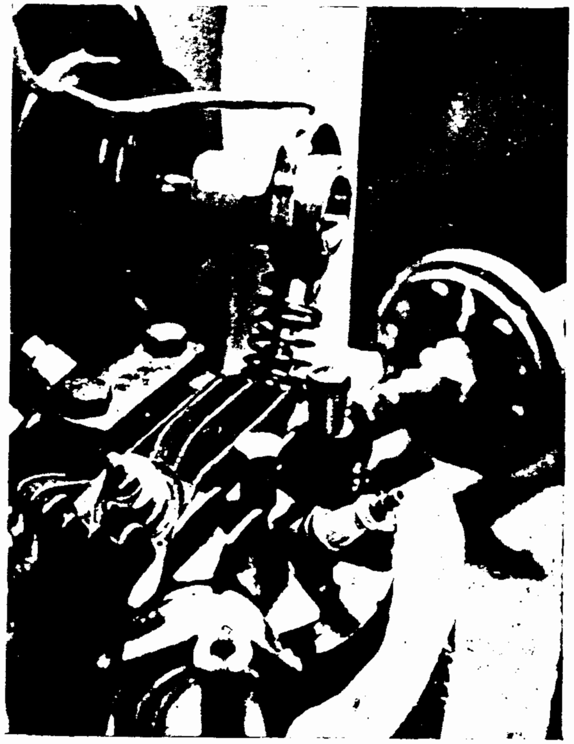


FIG. 4. Cam and Valve Arrangement.

4. EXPERIMENTAL EQUIPMENT

4.1. *Heat Transfer Measurements.*

A photograph of the equipment used for the heat transfer measurements is shown in Fig. 2 and the general layout is given as a line diagram in Fig. 3.

The equipment consisted of a horizontal brass pipe of 1″ inside diameter and 1¼″ outside diameter, through which compressed air was allowed to pass from the receiver of a two-stage reciprocating compressor. The mass flow of the air was controlled by the entrance gate valve, and measured by a calibrated orifice meter with D and D/2 taps made according to British Standard Specifications[22] though proportionally reduced. The pressure difference across the meter was measured by a vertical U-tube water manometer. To damp out the pulsations a large damping volume was introduced between the valve and the orifice meter.

The air was led into the heat transfer pipe through a petrol engine cylinder head. The air entered the cylinder head through the inlet port and past its valve and left it through the exhaust port the valve of which was removed, all other openings being sealed. The inlet valve was operated by means of a cam driven by a 220 V, 1 kw compound wound, D.C. motor of variable speed (see Fig. 4).

The frequency of pulsations could be varied between 5 and 40 c/s. by varying the speed of the motor which was measured by a Jaquet hand tachoscope with individual dials for time and for revolutions.

When tests were made under smooth flow conditions the poppet valve was kept open.

Downstream of the cylinder head a steam jacket consisting of a 2½″ G.I. pipe, suitably lagged, was fitted around the air pipe. A length of about 50″ was allowed between the valve and the steam jacket as a calming section while the effective heating length was 82¼″. Steam was generated at atmospheric pressure in a boiler consisting of a horizontal drum 15″ diameter and 18″ long fitted with three 1 kw, 220 V, immersion heaters and a gauge glass for noting the water-level. The condensate in the steam jacket was collected in a burette, with which the rate of condensation could be obtained.

The temperature of the air, before and after heating, were measured by mercury-in-glass thermometers. The thermometer bulbs were placed centrally in the air stream and the thermometer stems were isolated from the wall of the air pipe.

4.2. *Pressure Pulsations.*

Four types of pulsations were created by two different cams. Cam No. 1 (Fig. 5) closed the valve during 180° of each revolution and cam No. 2 (Fig. 6) during 280° of each revolution. For both cams, the base circle diameter was 2-7/16″, the thickness was ½″ and the lift 3/16″. With each of the cams two different types of pulsations were obtained (i) by leaving the end of the heat transfer pipe fully open, (ii) by attaching a ½″ orifice plate at the end. The four types of pulsations are designated by the symbols A, B, C and D, where A stands for

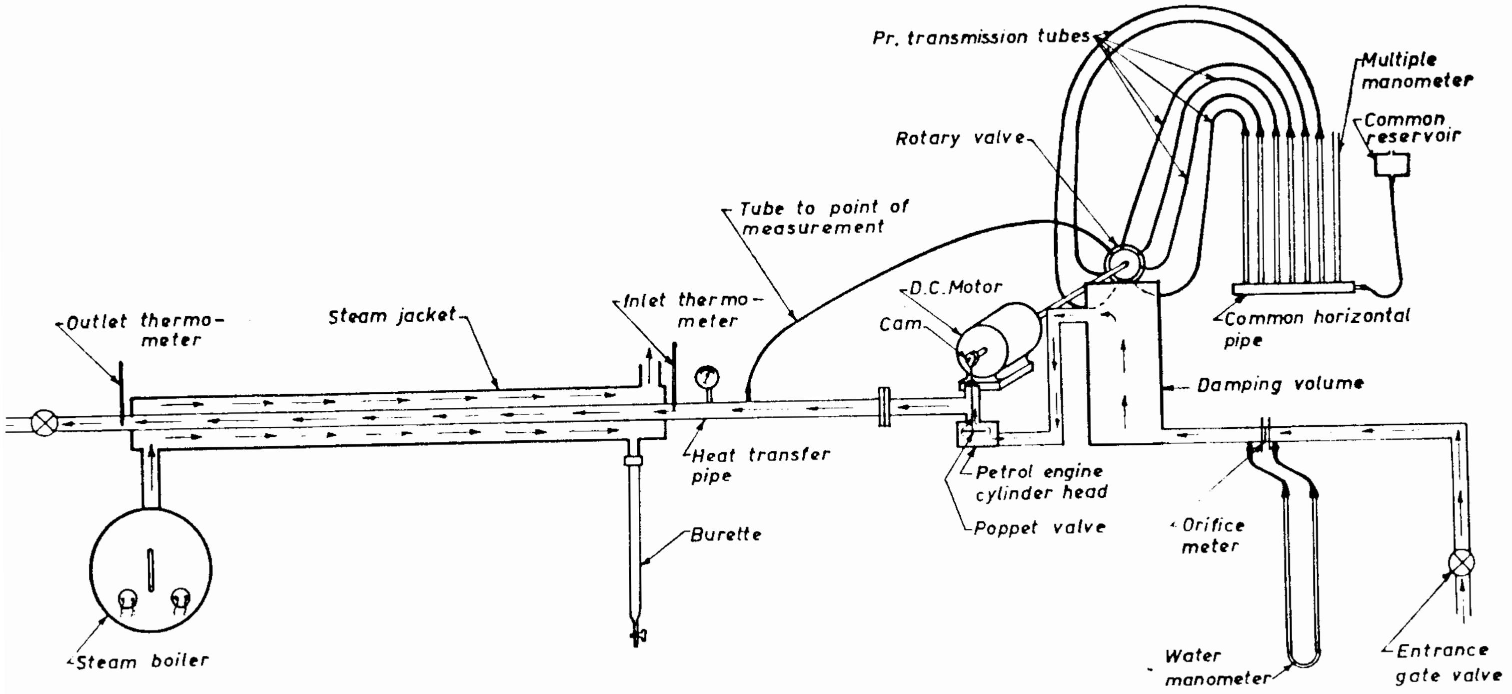


FIG. 3. Experimental Lay-out for Measurement of Heat Transfer to Pulsating Air.

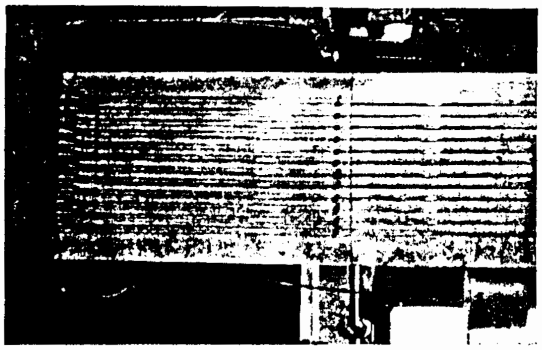


FIG. 9. Multiple Manometer.

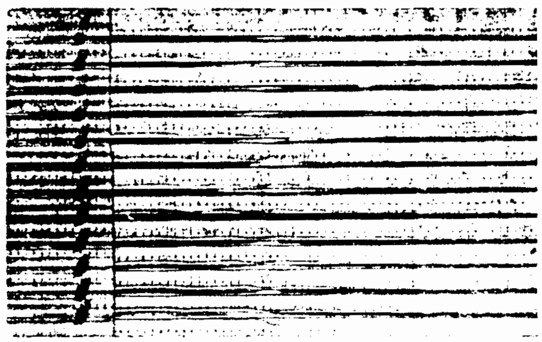


FIG. 10. A Typical Pressure Wave Record.

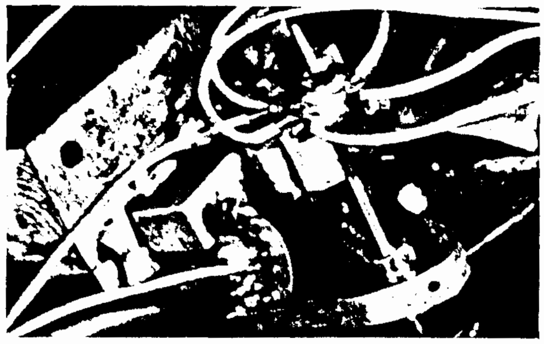
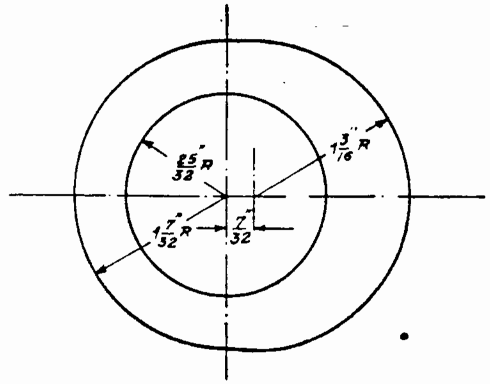


FIG. 7. A View of the Rotary Valve.

cam No. 1. B for cam No. 1 with orifice plate; C for cam No. 2, D for cam No. 2 with orifice plate. Of these A and C create progressive waves as the end of the tube is open to atmosphere. In the case of B and D standing waves are produced due to the reflection in part, from the orifice plate. The cams were rigidly mounted on the shaft of the motor by a sleeve and a key.



CAM 1

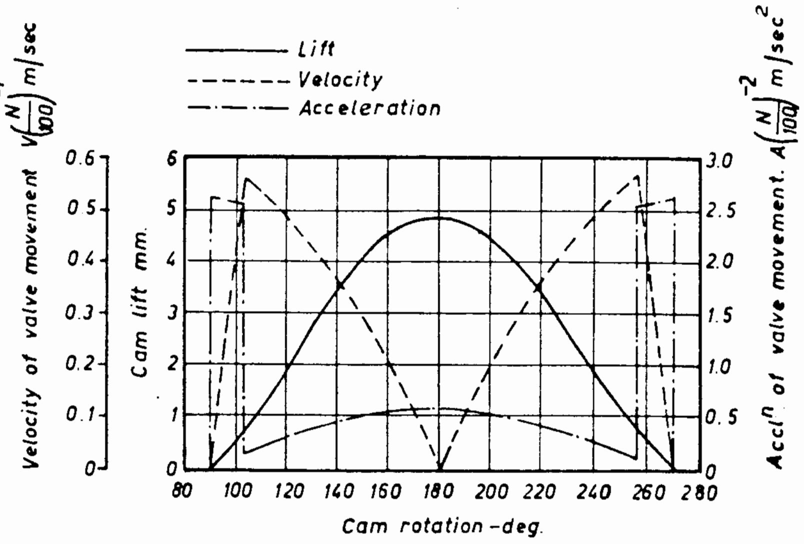
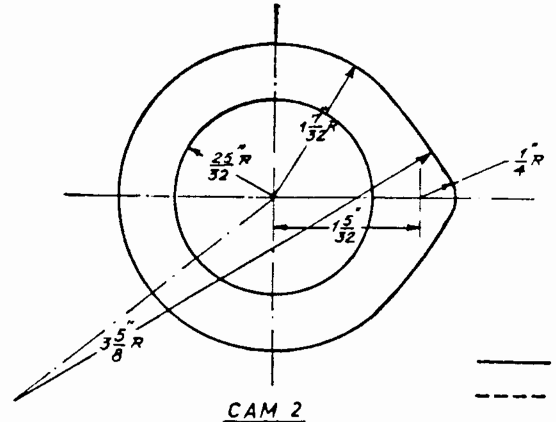


FIG. 5. Lift, Velocity and Acceleration Diagrams of Cam 1.



CAM 2

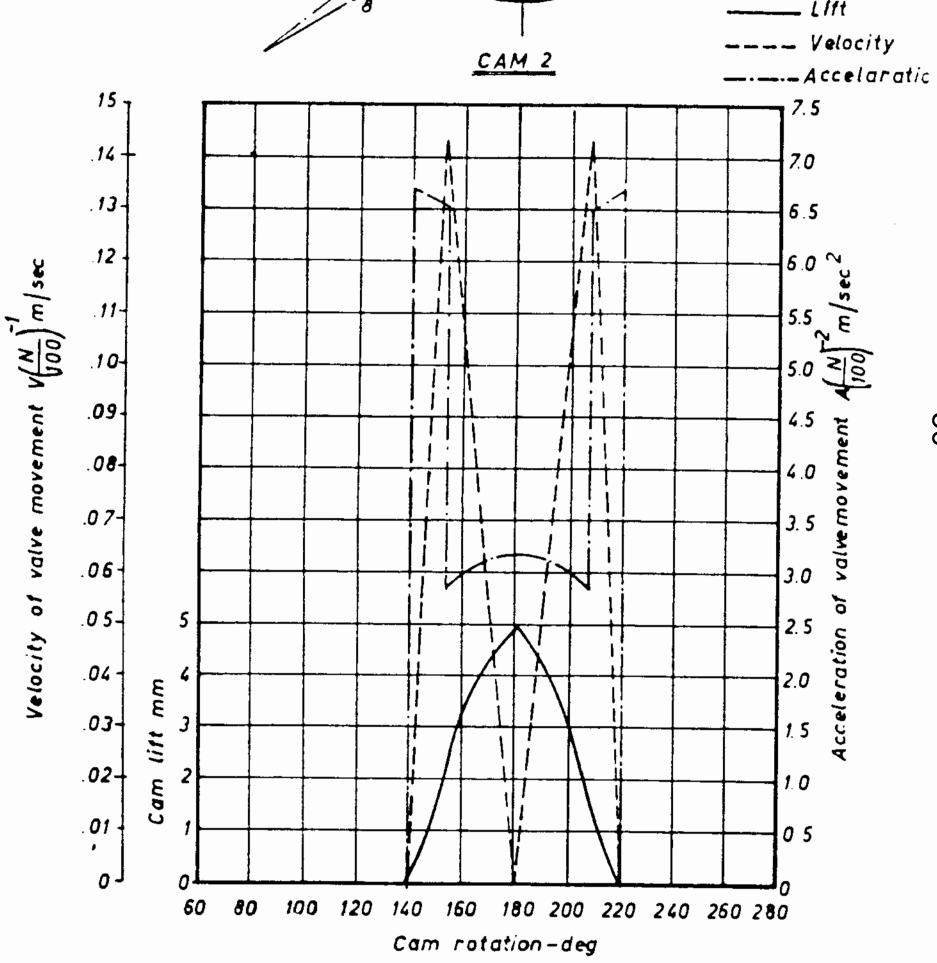


FIG. 6. Lift, Velocity and Acceleration Diagrams of Cam 2.

4.3. *Pressure Wave Measurements.*

An attempt was made to measure the pressure wave forms produced by these cams, by using a device based on a principle given by S c h w e i t z e r[23] and shown in Figs. 3, 7 and 8.

The instrument measures the pressures at any one instant in one manometer, the pressure at the next consecutive instant in another manometer, and so on. Thus

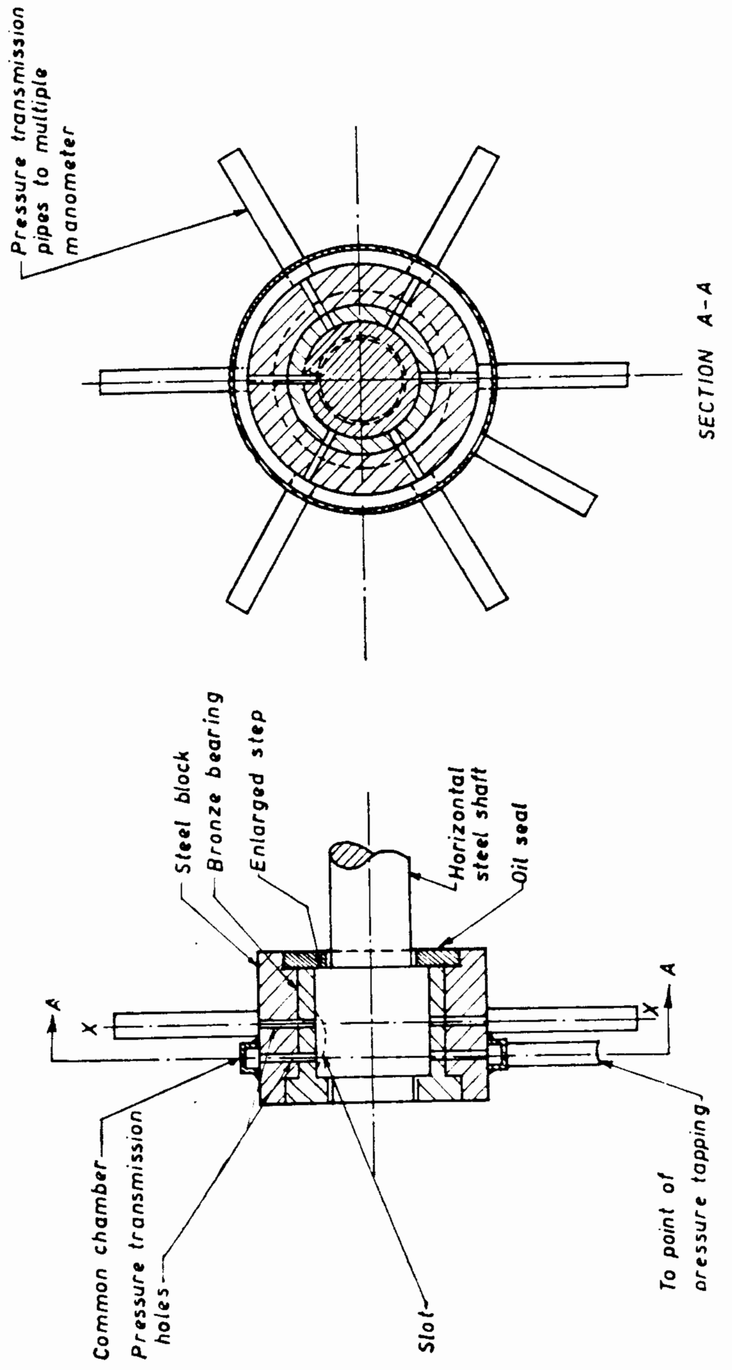


FIG. 8. Rotary Valve Indicator.

a series of manometers or other pressure measuring devices could give the pressures at any given point in the pipe at successive intervals of time.

The instrument essentially comprises a rotary valve and a sectioned drawing is given in Fig. 8. This consists of a horizontal steel shaft with an enlarged step, rotating in a bronze bearing housed in a steel block. Six pressure transmission holes, 1/16″ in diameter and equally spaced around the circumference are provided in each of the planes AA and XX in such a manner that each of the holes in one plane is exactly opposite to a corresponding hole in the second plane. Each hole in plane XX is connected to a separate manometer by the pressure transmission pipe while all the holes in plane AA lead to a common chamber which is connected to the point at which the pressure wave form has to be determined. A slot in the shape of a feather key-way is provided on the machined step in such a way that if the shaft is rotated corresponding holes in the two planes are connected at successive intervals of time.

Air-tightness was ensured by the oil film in the bearing since the oil-seals were sufficient for the pressures encountered. The shaft of the rotary valve was directly connected to the shaft operating the cam.

The rotary valve was connected to a multiple manometer (Fig. 3 and Fig. 9) in the manner commonly used in wind tunnel testing.[24] The manometer tubes were filled with water coloured blue with methylene dye and a few drops of photographic wetting agent were added to lower the surface tension.

The readings obtained on the multiple manometer with the help of the rotary valve mechanism were not representative of the actual static pressure in the pipe at the point of the pressure tapping due to the interaction of the intervening air space.* Pressures measured at the entrance of the heat transfer section, however, would allow to correlate the heat transfer measurements with the steepness of the differently shaped wave forms (see Fig. 10).

5. EXPERIMENTAL TECHNIQUE

The experimental set-up was designed to allow the determination of the total quantity of heat transferred through the wall of a horizontal tube to pulsating air from the surrounding steam jacket under different conditions of air mass flow, frequency of pulsations, pressure amplitude and wave form. The major quantities measured were therefore air and condensate flows, the temperature of the steam in the jacket and of the air before and after heating and the frequencies of the pressure waves, for different shapes and amplitudes.

First, preliminary experiments were conducted to ascertain the accuracy of and to calibrate, the various measuring arrangements that were used in the final experiments.

* Accurate measurements of pressure and velocity of oscillations of large amplitudes induced on the air in a long pipe by an oscillating piston have been undertaken by Lettau.[25]

The boiler was the first part of the apparatus to be switched on. The steadiness of the conditions in the heat exchanger was judged from the rate of condensate formation. As soon as steady temperature conditions were achieved, the valve imparting the pulsations was kept in the open position and compressed air was allowed to pass through the tube. The rate of flow was adjusted to the required value by regulating the entrance gate valve. When the temperatures of the air at inlet and outlet and the condensate formation became steady, readings were taken. From the measurements, the rate of heat transfer for steady flow conditions was calculated.

The motor to operate the valve was then started and set at its lowest speed. This changed the pressure difference across the orifice plate, which was restored to its original value by an adjustment of the entrance gate valve. When a steady state was again attained, readings were taken as under steady flow conditions. In addition, the motor speed was measured, which represented the frequency of pulsations.

The mass flow was kept constant and the rate of heat transfer was calculated for various frequencies. This was repeated at different mass flows for both the cams. Several sets of readings taken at different times showed very little discrepancy.

TABLE II

Heat Transfer with Steady Flow

Sl. No.	Re	Nu (Actual)	Nu (Theoretical)	Deviation %
1	9,210	24·16	30·72	– 21·4
2	10,310	27·24	33·65	– 19·0
3	11,770	30·01	37·41	– 19·8
4	13,620	33·62	41·98	– 19·9
5	14,280	35·18	43·65	– 19·4
6	14,800	35·54	44·87	– 20·8
7	16,090	38·52	47·97	– 19·7
8	17,890	42·10	52·24	– 19·4
9	19,640	45·12	56·36	– 19·9
10	21,500	48·15	60·53	– 20·5
11	22,380	49·75	62·52	– 20·4
12	23,450	51·65	64·86	– 20·4
13	26,330	54·64	71·12	– 23·2
14	27,820	58·70	74·30	– 21·0
15	29,400	61·85	77·80	– 20·5
16	30,410	63·30	79·80	– 20·7
			Mean Deviation %	– 20·4

6. EXPERIMENTAL RESULTS

Experimental data obtained in the manner described were evaluated and calculated as outlined in the Appendix. The following results were obtained:—

6.1. *Heat Transfer with Steady Smooth Flow.*

The data for steady smooth flow are evaluated and shown in Table II and plotted in Fig. 11. They may be represented by the equation:

$$Nu = \cdot 0191\ Re^{\cdot 785}$$

In general these values lie an average of 20·4% below the Dittus-Boelter[26] line:

$$Nu = \cdot 0207\ Re^{\cdot 8}$$

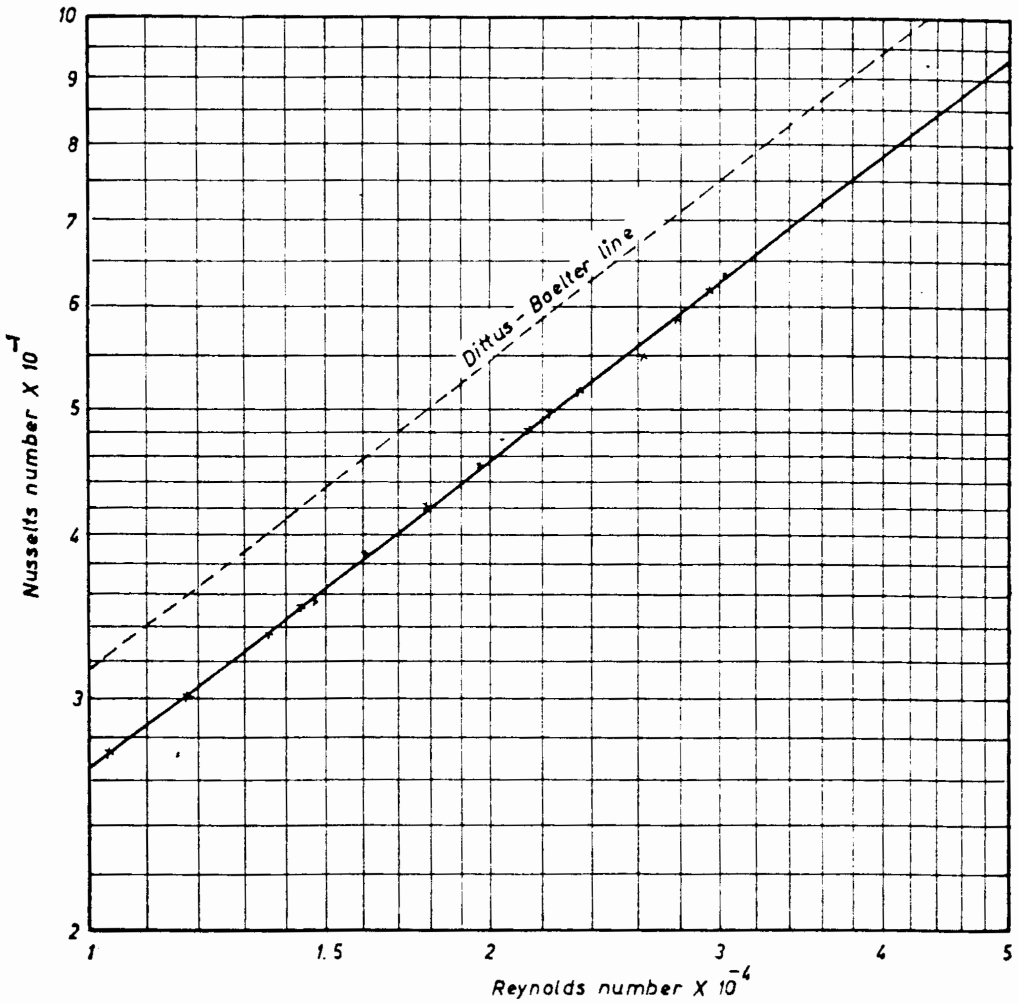


FIG. 11. Re *vs.* Nu for Steady Smooth Flow.

This is probably due to uneven heating of the heat transfer pipe along its length, the temperature being lower near the inlet end and higher at the outlet end. The tendency of these results, however, has no significance for the problem investigated here since the heat transfer coefficients with pulsating flow were compared with the coefficients of smooth flow under otherwise identical conditions.

6.2. *Heat Transfer with Pulsating Flow.*

The results obtained from the heat transfer experiments with pulsating flow are shown in Figs. 12 and 13.

With wave shape 'A', the ranges investigated were: $6 < n < 39$ and $6{,}400 < Re < 25{,}060$. The maximum reduction recorded, of the Nusselt number was about 43% at $n = 22 \cdot 5$ and $Re = 6{,}400$. The maximum increase was about 21% at $n = 35 \cdot 5$ and $Re = 20\,060$. In general, the Nusselt numbers for Reynolds numbers $> 10{,}000$, were less than the steady state below a frequency of about 31 to 32 c/s. and greater above this value reaching the maximum at a frequency of about 36 c/s.

With wave shape 'B', the ranges investigated were $8 < n < 40$ and $6{,}400 < Re < 25{,}060$. The maximum decrease was about 20% at $n = 8$ and $Re = 14{,}970$. The maximum increase was about 14% at $n = 27$ and $Re = 6{,}400$.

With wave shape 'C', the ranges investigated were $9 < n < 26$ and $6{,}400 < Re < 14{,}970$. The Nusselt numbers were always less than those for steady flow, the minimum decrease being about 6% at $n = 15$ and $Re = 10{,}270$.

With wave shape 'D', the ranges investigated were $9 < n < 26$ and $6{,}400 < Re < 12{,}510$. The Nusselt numbers were always higher than those for steady flow within the ranges considered with a maximum improvement of about 42% at $n = 15 \cdot 5$ and $Re = 6{,}400$.

Fig. 12 shows typical curves of the Nusselt numbers against frequency obtained for all observed wave shapes and investigated Reynolds numbers. The Nusselt value for steady flow is also indicated in each figure. The same data are plotted in Fig. 13 in the form of the relative change of Nusselt number depending on frequency.

6.3. *Comments.*

It is seen from Figs. 12 and 13 that all experimentally obtained results do not fall on a smooth line though the trend of the interdependence is unmistakable. This emphasises the unsteady character of the process of heat transfer when it occurs in pulsating flow. A similar phenomenon has been noted by Schultz-Grunow[27] in his experiments on the velocity profile in pulsating water flow where the readings of the velocity at different points across the section of the pipe do not fall along a smooth curve. Yet the results obtained appear to be dependable in spite of the scatter since approximately the same curves were obtained by experiments repeated under the same conditions. The experiments with superimposed standing wave systems 'B' and 'D' show the same characteristics although in an accentuated manner.

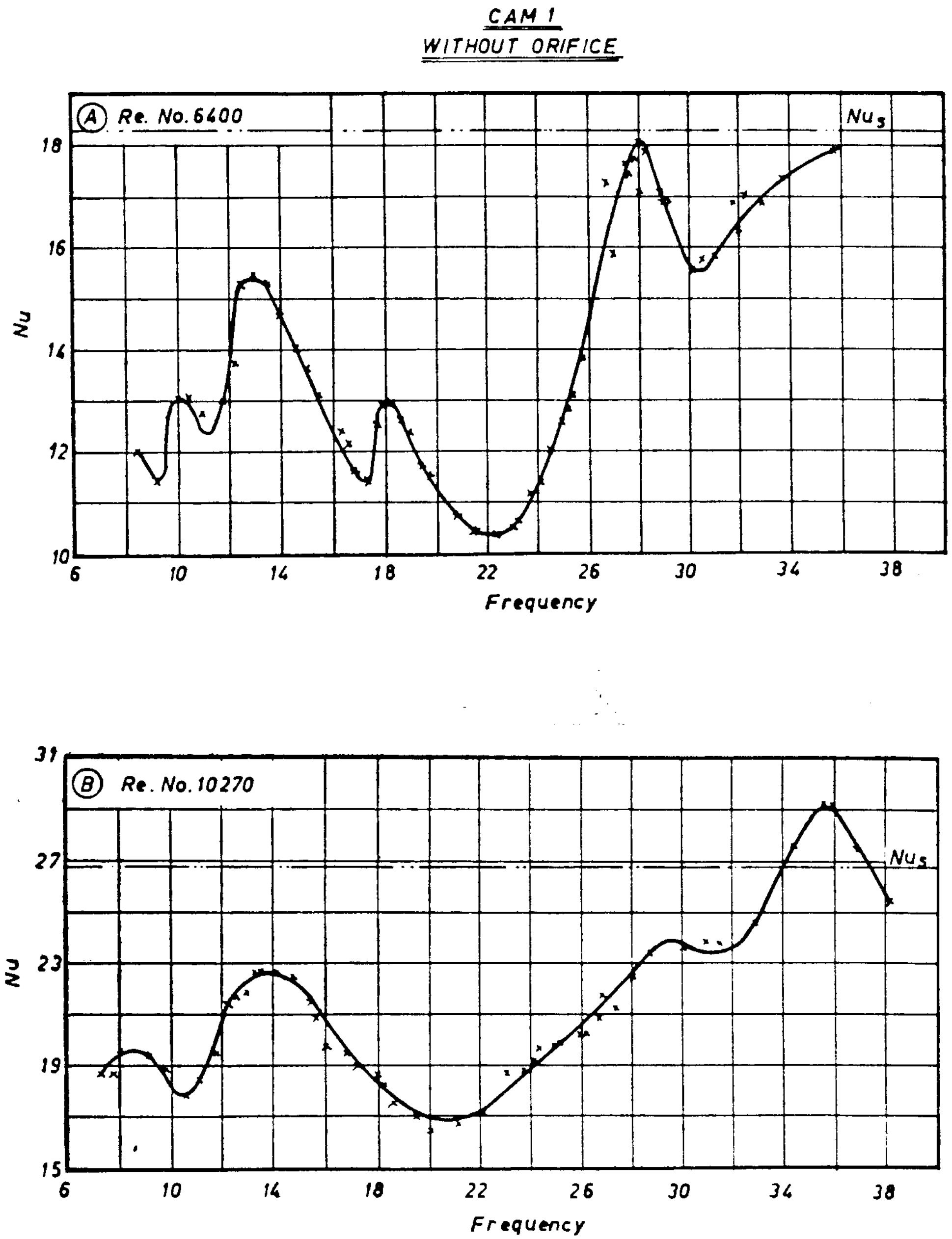


FIG. 12 (A, B)

6.4. *Conclusions.*

From the results of the experiments reported above it is possible to conclude that the process of h. .t transfer to a pulsating medium may be considerably modified

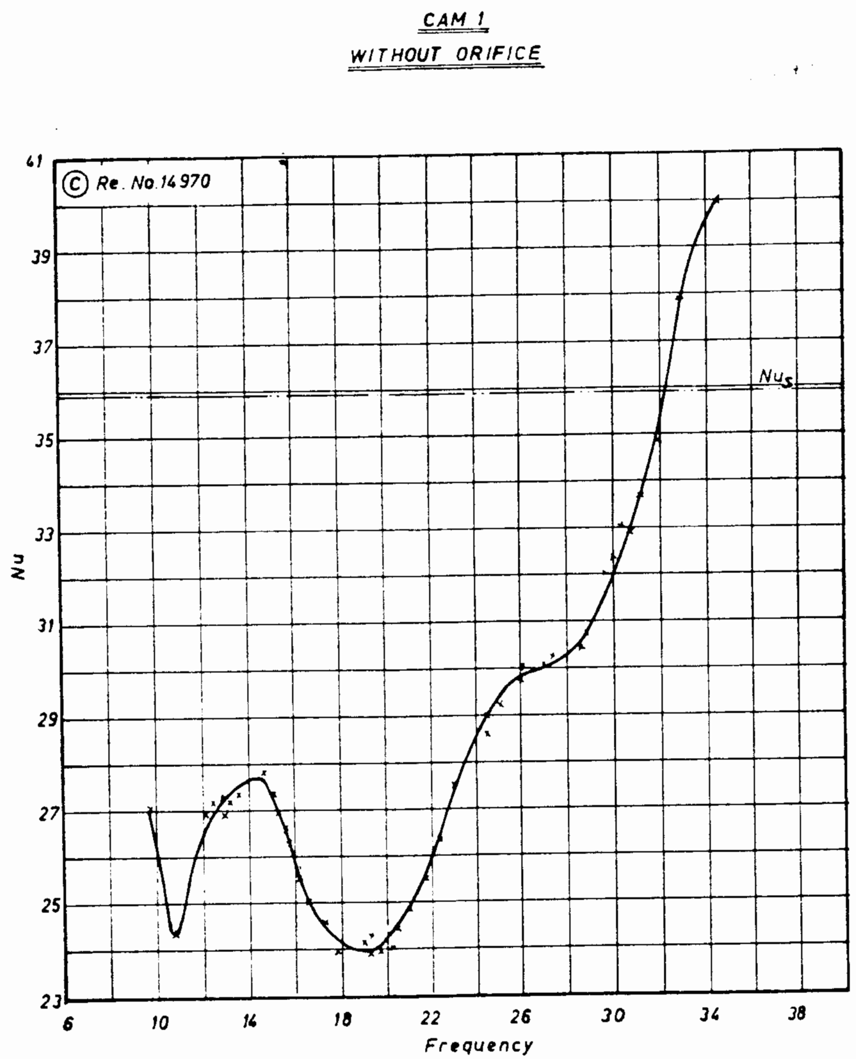


Fig. 12 (C)

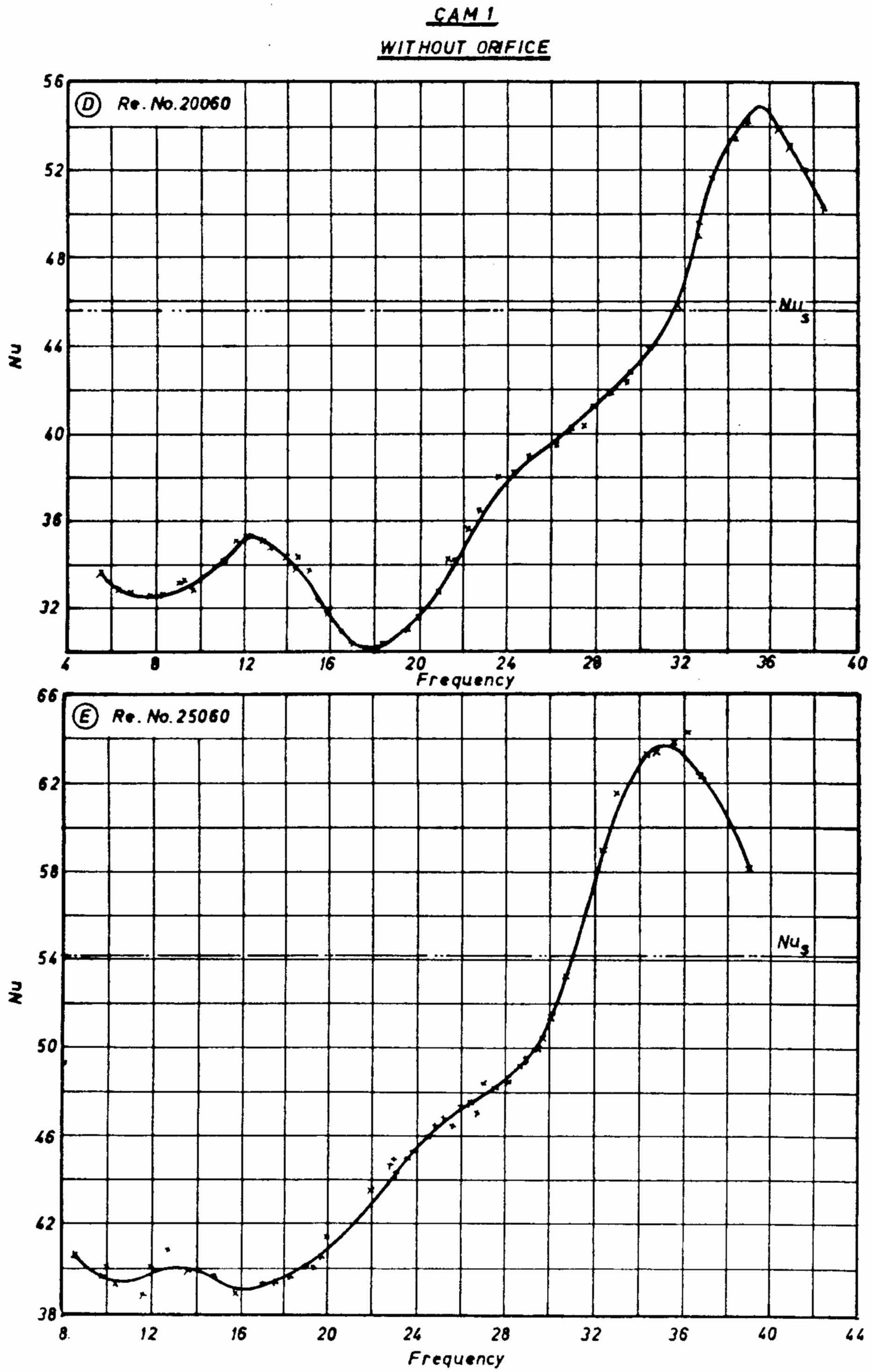


Fig. 12 (D, E)

by the nature and frequency of pulsations imposed on the medium. The change of the rate of heat transfer depends on the frequency of the pulsations and the Reynolds number whereby depending on them, negative or positive changes are encountered.

The onset of improved heat transfer occurs across a comparatively narrow band of frequencies and is maintained for a small range only for a cam which opens the passage for nearly half a revolution (see Fig. 13 A). For a cam opening only

CAM 1
WITH ORIFICE

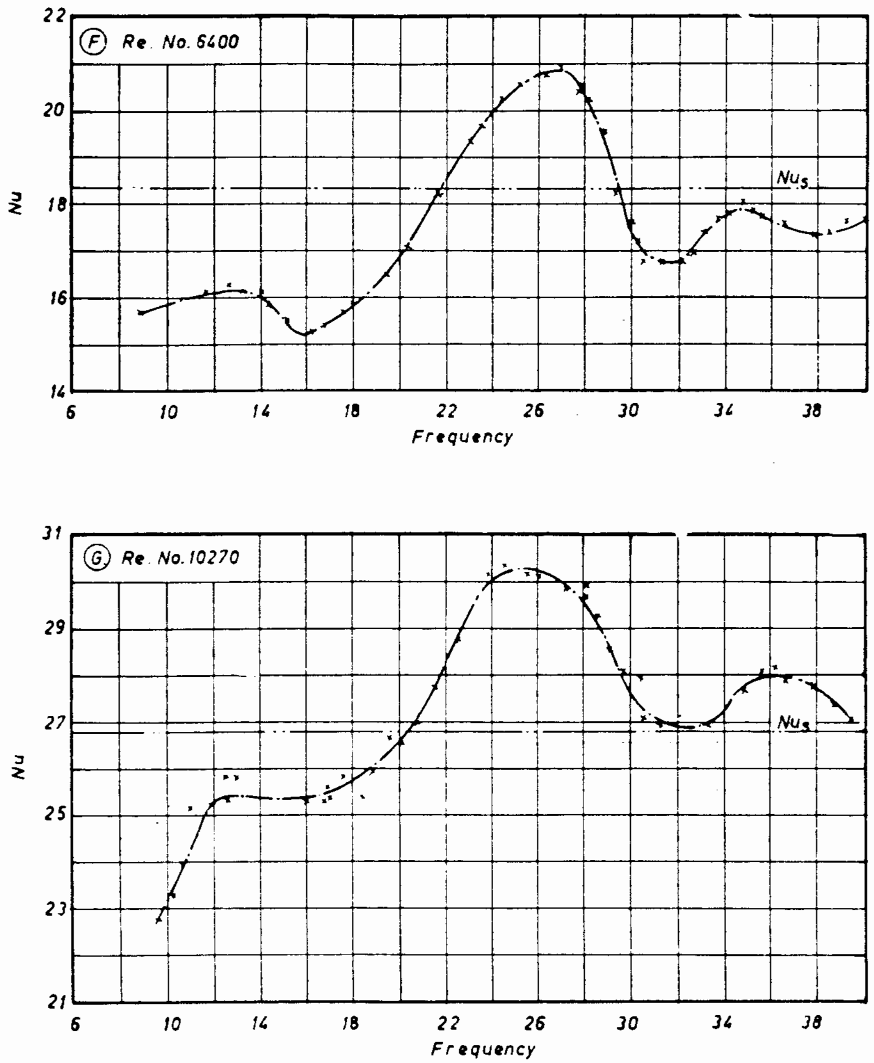


FIG. 12 (F, G)

CAM 1
WITH ORIFICE

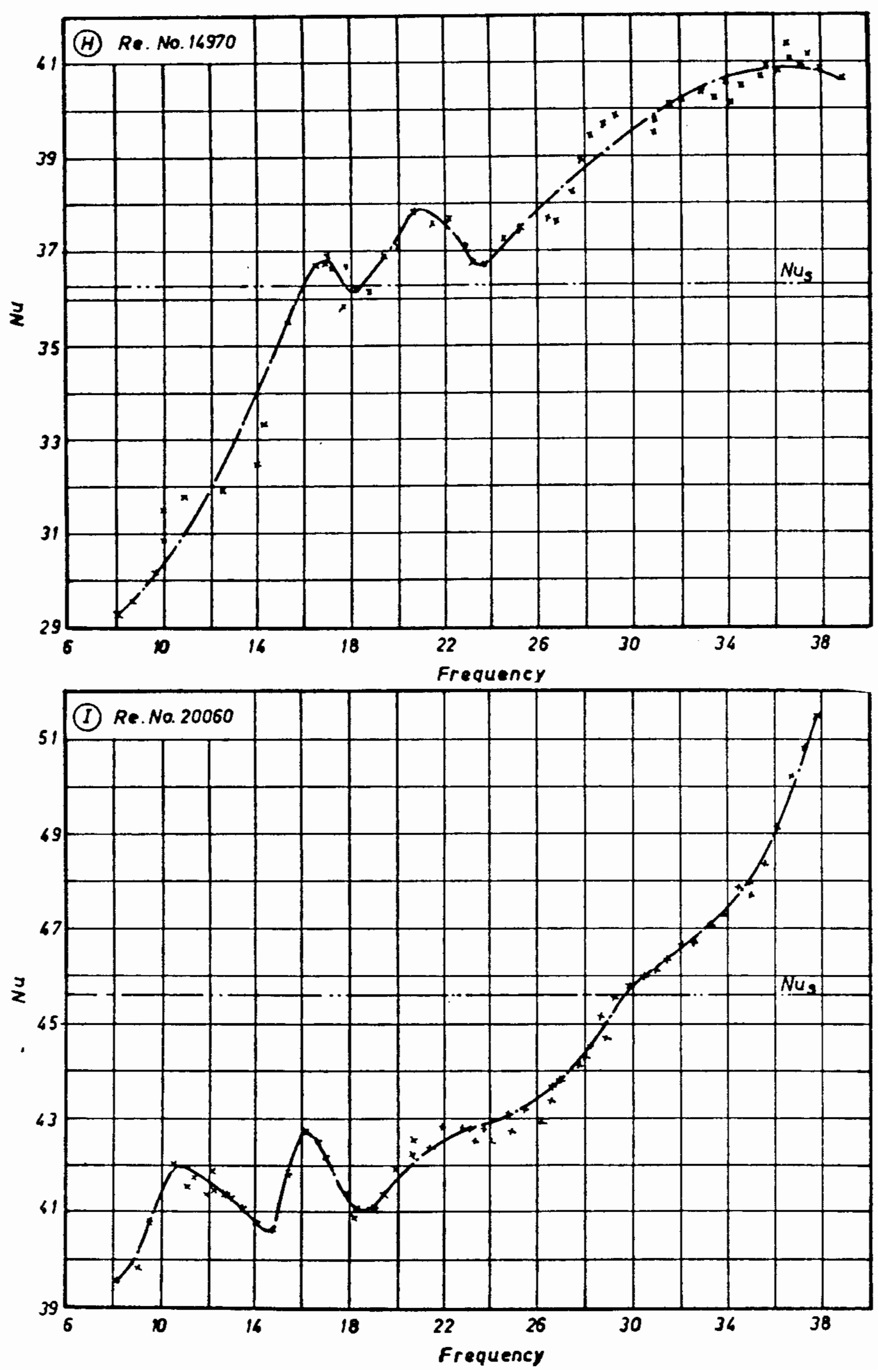
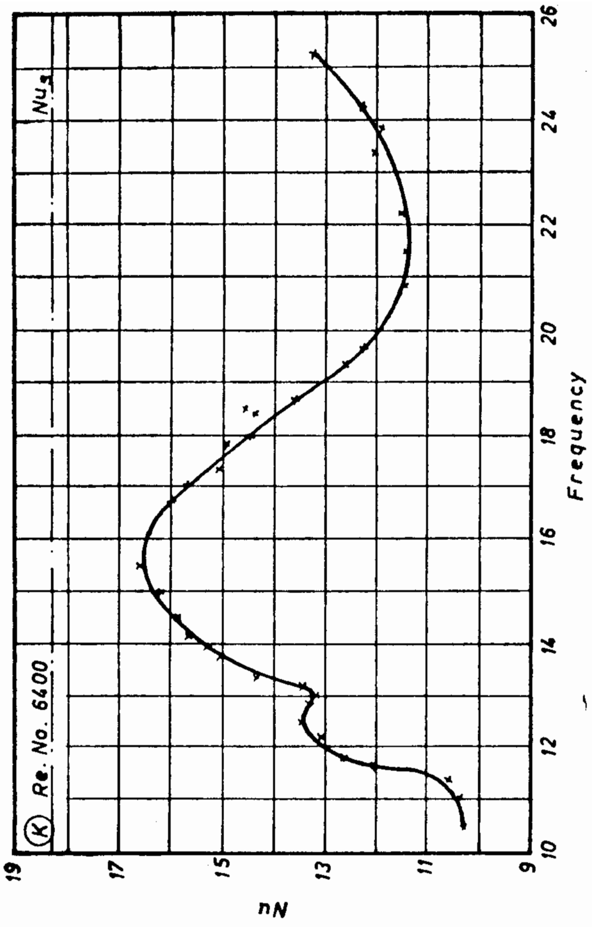


FIG. 12 (H, I)

CAM 2
WITHOUT ORIFICE



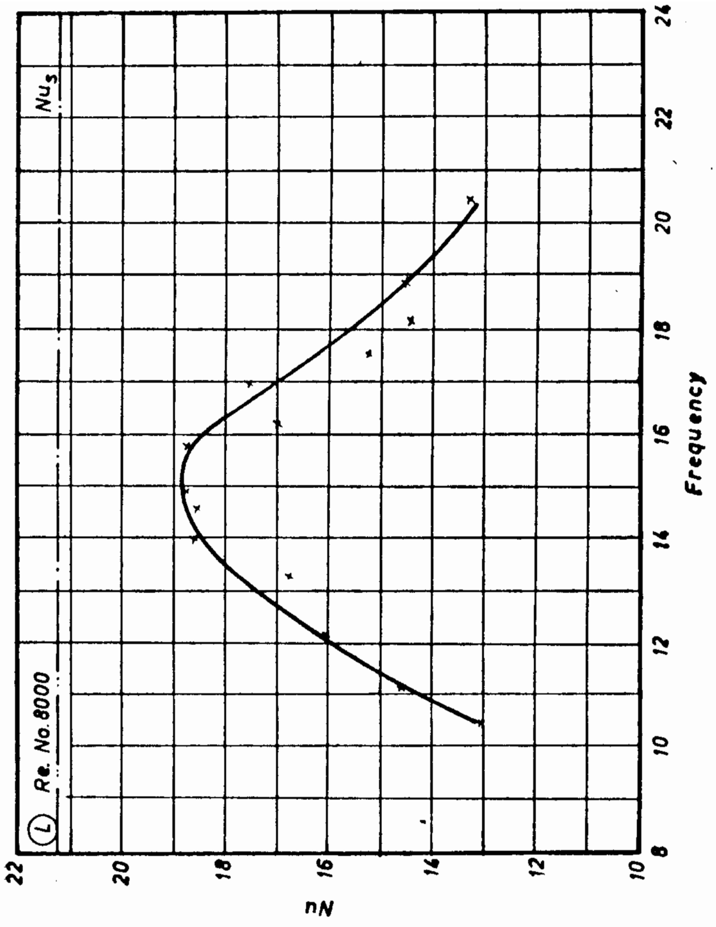


FIG. 12 (K, L)

CAM 1
WITH ORIFICE

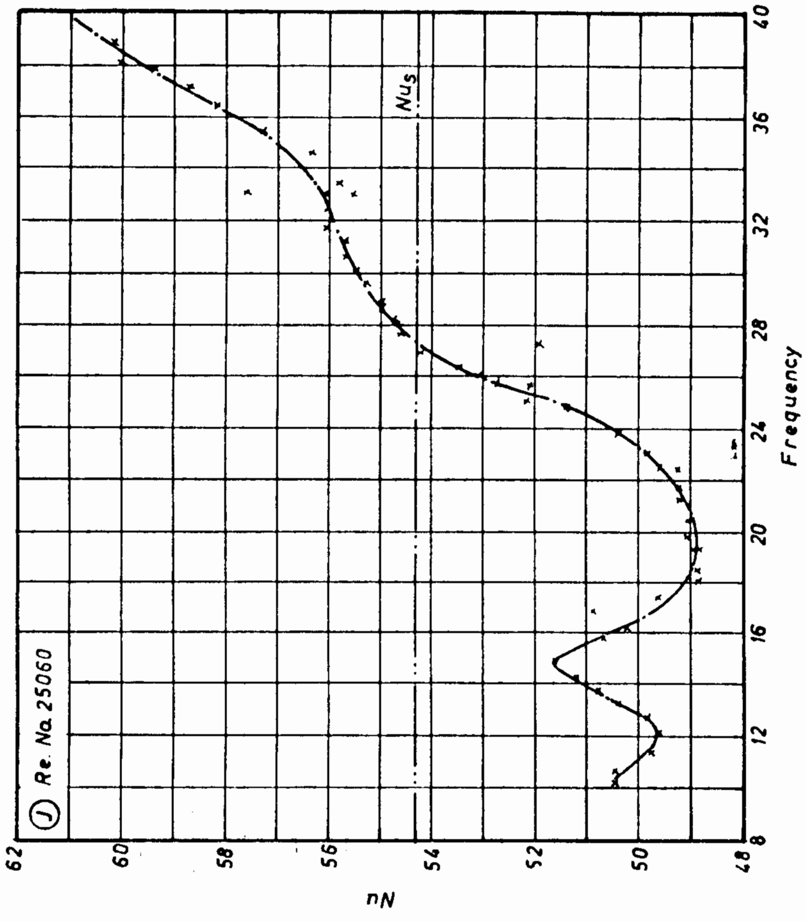


FIG. 12 (J)

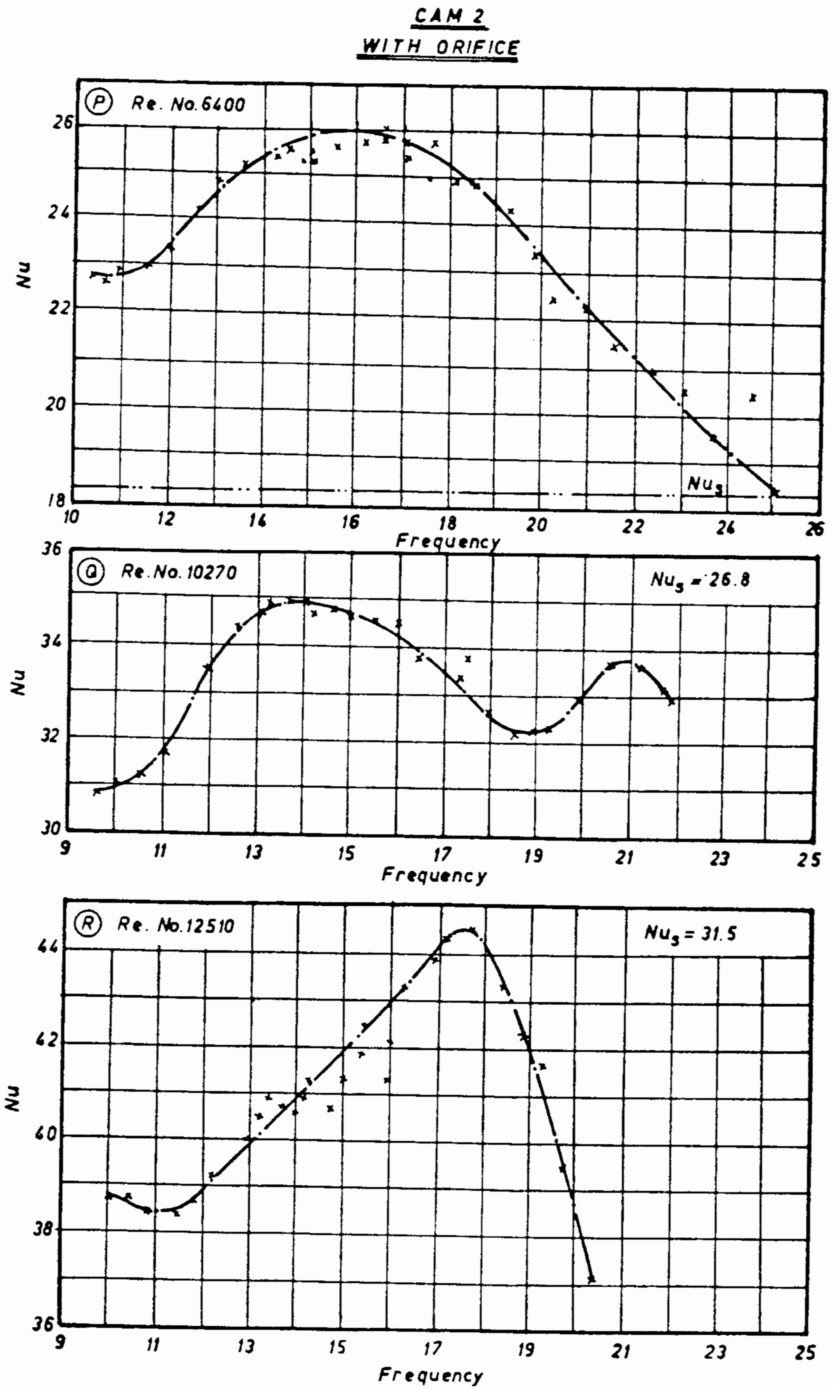


FIG. 12 (P, Q, R)

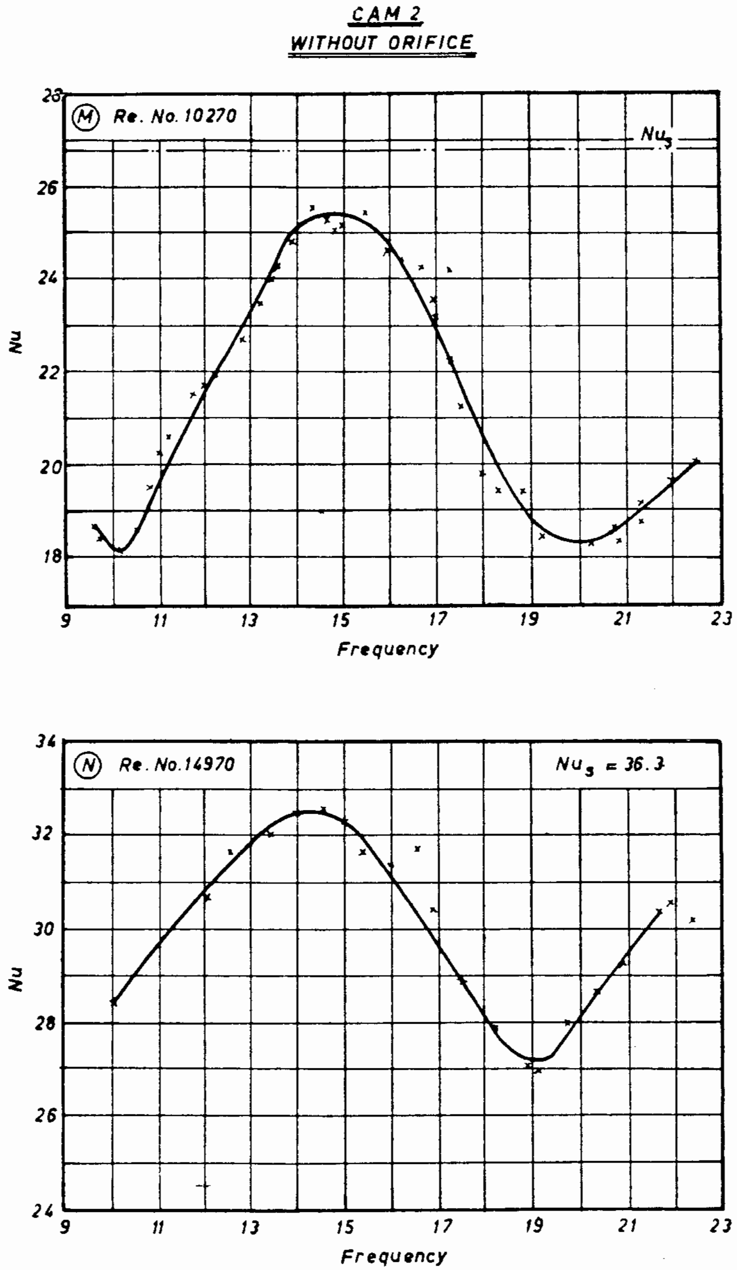


FIG. 12 (M, N)

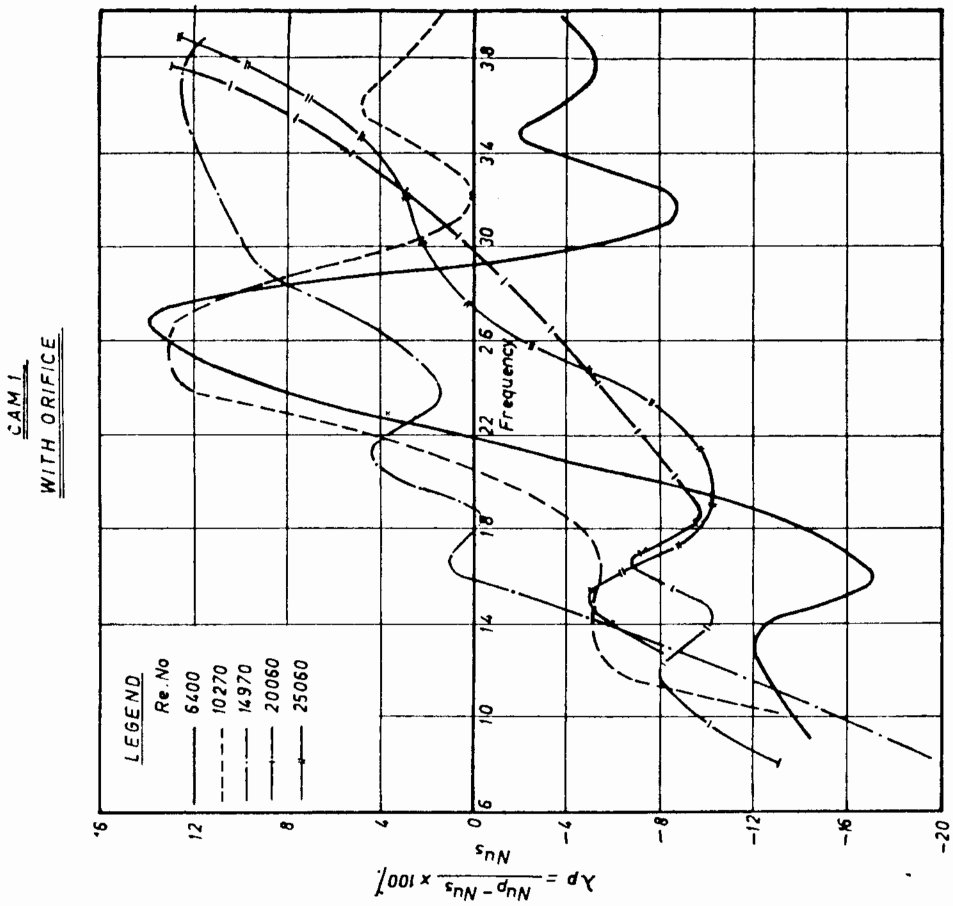


FIG. 13 B

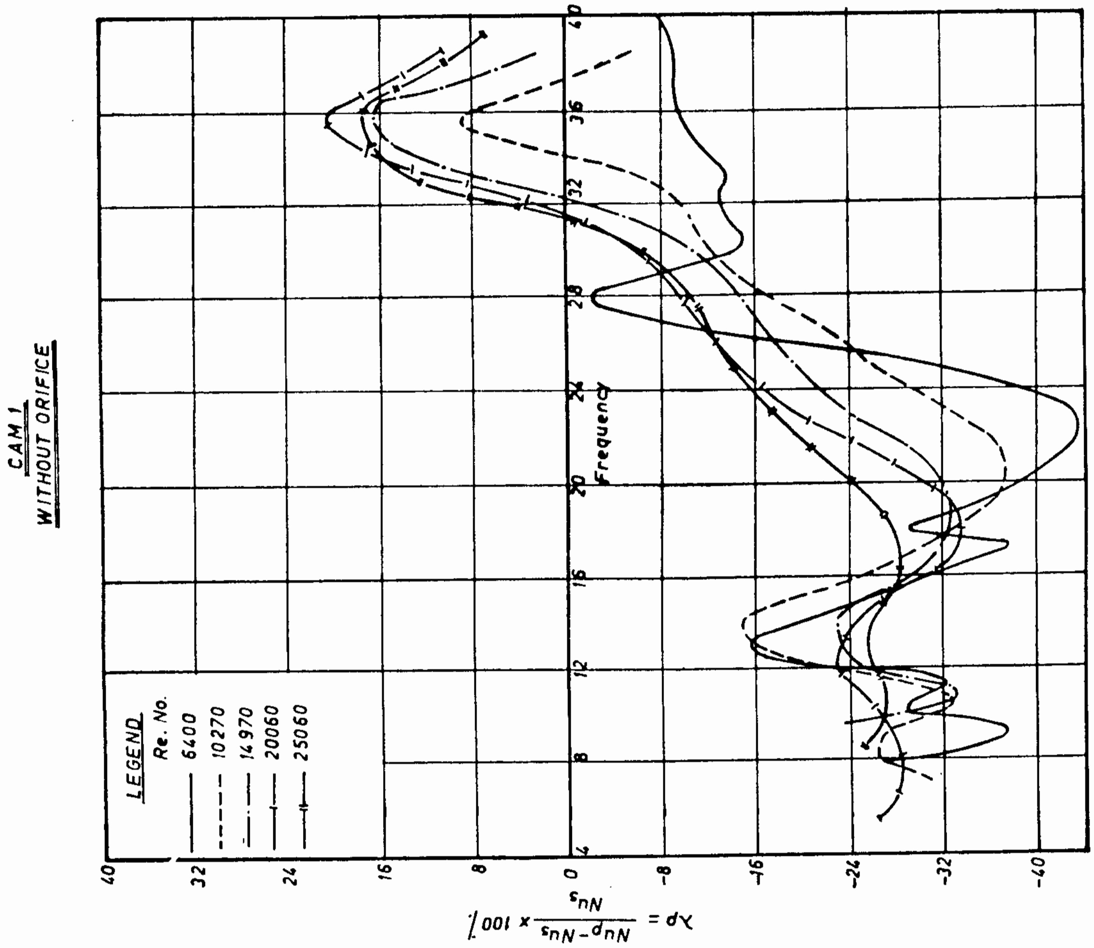


FIG. 13 A

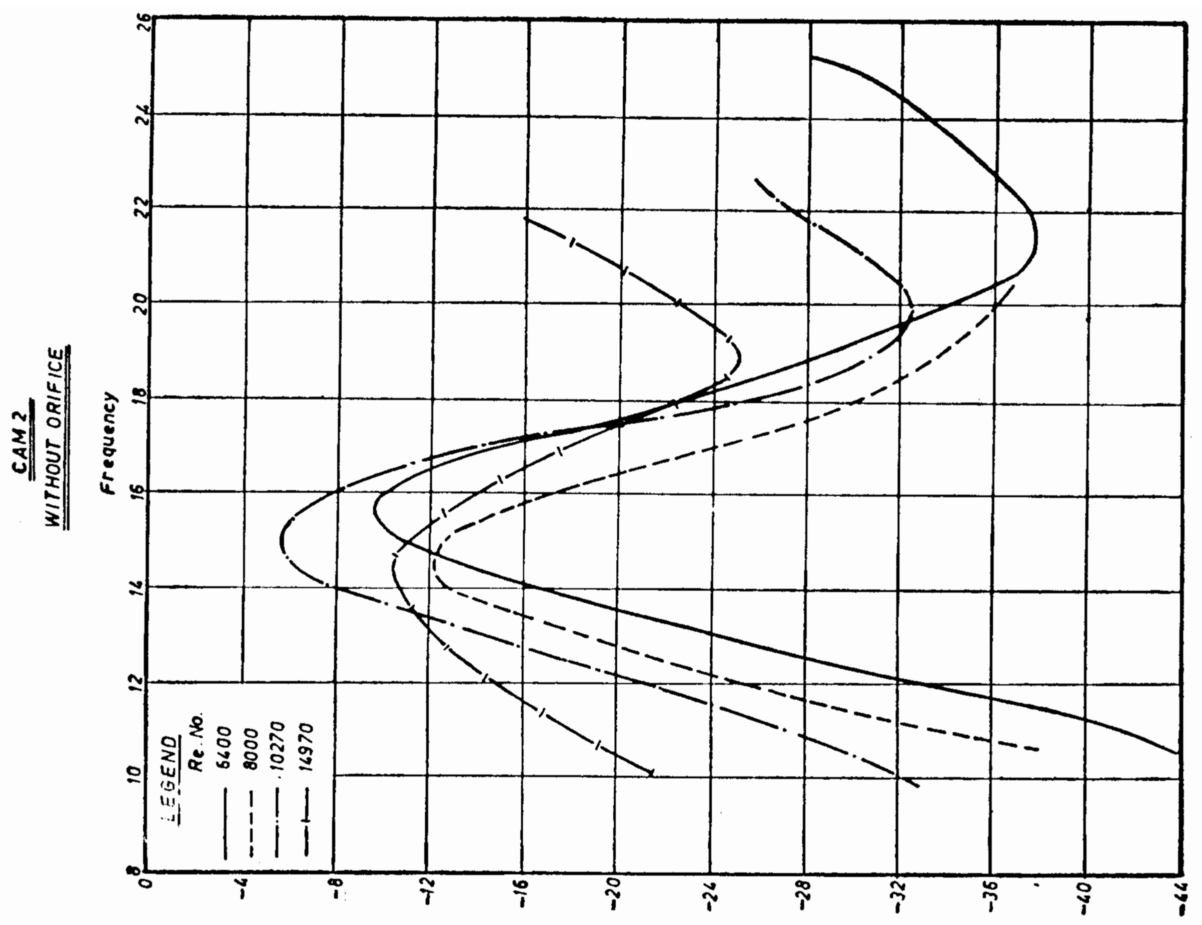


FIG. 13 C

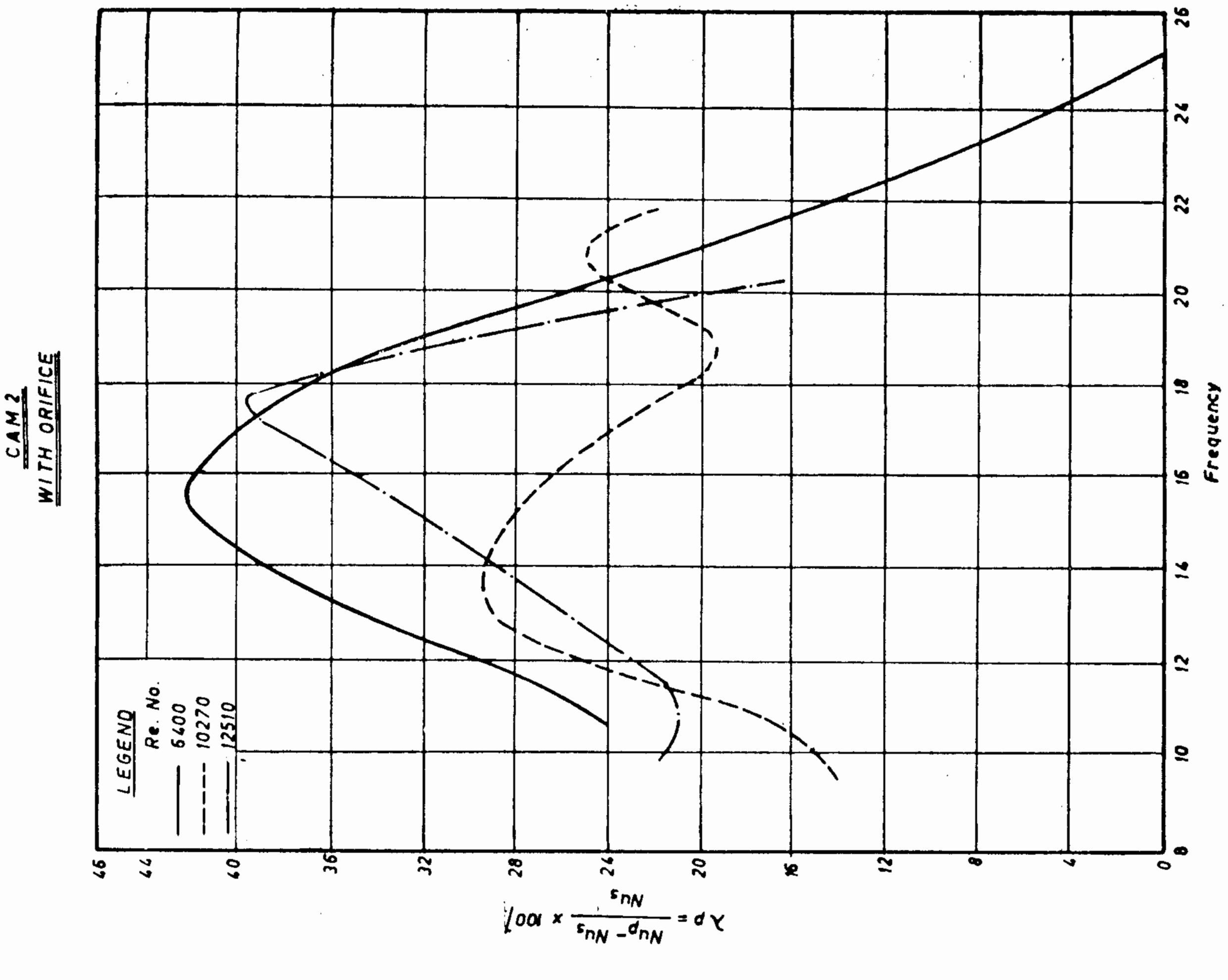


FIG. 13 D

for 80° of cam rotation, no improvement over steady flow heat transfer is obtained (see Fig. 13 C). However, by fitting an orifice plate reducing the free exit area of the heat transfer pipe the first mentioned cam achieves a general improvement above a lower frequency and over a larger range of frequencies (Fig. 13 B). The operation of the second cam results in a general increase of rates of heat transfer for all frequencies investigated (Fig. 13 D).

7. Acknowledgements

This work was initiated by the first author and undertaken at the Department of Internal Combustion Engineering, Indian Institute of Science, Bangalore, as part of the scheme: ' Hot Air Engines and Their Development ' under the auspices of the Council of Scientific and Industrial Research, New Delhi. The authors are grateful to both these bodies for the facilities which have been placed at their disposal.

Thanks are due to Mr. N. G. S. Prasanna, Statistical Assistant, for the laborious numerical work required to evaluate the results.

8. List of Symbols and Units, Figures and Tables

8.1. *List of Symbols and Units.*

Symbol	Significance	Units
A	Area of heat transfer tube	m^2
A_1	Area of orifice	m^2
C_d	Coefficient of discharge	..
C_p	Mean specific heat of air at constant pressure	cal./gm. °C.
c/s	Cycles per second	..
D	Inner diameter of heat transfer pipe	m
D_1	Diameter of orifice	m
D_o	Outer diameter of heat transfer section	m
E	$1/\sqrt{(1 - m^2)}$, where $m = D_1{}^2/D^2$	..
g	Gravitational constant	m/s^2
h	Differential head across orifice plate	m of water
H	Rate of heat transfer	cal/s
k	Thermal conductivity	cal/s (m^2) (° C. per m)
L	Effective heating length of heat transfer section	m
n	Frequency	c/s
Nu	Nusselt number	..
Nu_p	Nusselt number for pulsating flow	..
Nu_s	Nusselt number for steady flow	..
Re	Reynolds number	..
T_i	Temperature at inlet to heat transfer section	° C.
T_o	Temperature at outlet of heat transfer section	° C.
T_s	Temperature of formation of steam at atmospheric pressure	° C.
Δt_m	Logarithmic mean temperature	° C.
Δt_{min}	$T_s - T_o$	° C.
Δt_{max}	$T_s - T_i$	° C.
U	Overall coefficient of heat transfer related to outer area of pipe within the heat transfer section	cal./s.m^2.°C.
V	Velocity of air in heat transfer section	m/s
Q	Rate of flow of air	m^3/s
W	Mass rate of flow of air	g/s
ρ	Specific gravity of air relative to water at the temperature and pressure up-stream of orifice	..
ν	Kinematic viscosity of air	m^2/s
λ_p	$\dfrac{Nu_p - Nu_s}{Nu_s} \times 100$	%

8.2. *List of Figures.*

8.3. *List of Tables.*

9. LIST OF REFERENCES

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10. APPENDIX

Experimental Evaluation

The volume of air flowing through the pipe was calculated from the formula:

$$Q = C_d . A_1 . E . \sqrt{2g}\ \sqrt{h/\rho}.$$

The velocity in the pipe was then obtained from the formula:

$$V = Q/\pi\ (D^2/4)\ .\text{m/s}.$$

From this the Reynolds number was calculated:

$$Re = VD/\nu.$$

The Reynolds numbers for pulsating flow and for steady flow were calculated in a similar manner in accordance with Martinelli *et al.*[3] Actual numerical calculations showed that for the amount of variation of temperature during experiments with a given mass flow the variation of velocity due to temperature changes was in general negligible, the error in the density being less than 2%.

For calculating the heat transfer coefficient the logarithmic mean temperature differences were used:

$$\Delta t_m = (\Delta t_{max} - \Delta t_{min})\ /\log.\ \Delta t_{max}/\Delta t_{min}.$$

The amount of heat transferred to the air in the heat transfer section is given by

$$H = U . \pi D_0 . L . \Delta t_m,$$

where U = overall coefficient of heat transfer related to outer area of pipe within the heat transfer section.

The amount of heat absorbed by the air is

$$H = C_p\ W(T_0 - T_i).$$

Equating the two, the heat transfer coefficient is obtained as:

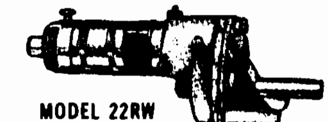
$$U = C_p\ W(T_0 - T_i)/\pi\ D_0\ L(\Delta t_m).$$

From this the Nusselt number $Nu = UD_0/K$ is obtained.

Nusselt numbers were calculated for steady and pulsating flows at the several Reynolds numbers, giving Nu_s and Nu_p respectively. The percentage difference $(Nu_p - Nu_s)/Nu_s . 100\%$ were then obtained.

AIR MOTORS

A R O



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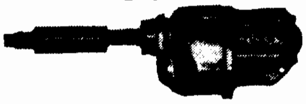
CLECO

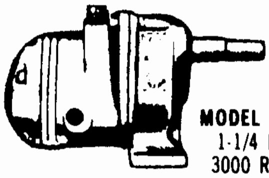
MODELS
IONS-12A & IOD-12
1000-1500 RPM
List Price $325
SPECIAL REBUILT PRICE **$65**

CLECO



MODEL 101-NS
495 RPM
List Price $605
SPECIAL REBUILT PRICE **$105**

CHICAGO PNEUMATIC



MODEL 3007
1-1/4 HP
3000 RPM
List Price $560
SPECIAL REBUILT PRICE **$155**

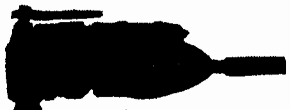
INGERSOLL-RAND

MODEL 380M
1100 RPM
List Price $1300
SPECIAL REBUILT PRICE **$295**

INGERSOLL-RAND

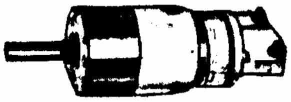
MODEL 180
1000 RPM
List Price $695
SPECIAL REBUILT PRICE **$190**

INGERSOLL-RAND



MODEL 30
185 RPM
List Price $5600
SPECIAL REBUILT PRICE **$1200**

KELLER

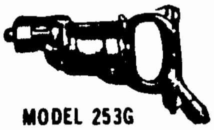


MODEL 1001, IVR-42N
325 RPM
REVERSIBLE
List Price $550
SPECIAL REBUILT PRICE **$195**

KELLER

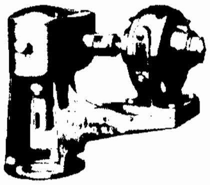
MODEL 16 J44
560 RPM
List Price $685
SPECIAL REBUILT PRICE **$155**

THOR



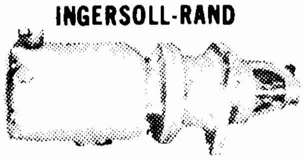
MODEL 253G
7/16'' Capacity 700 RPM
List Price $640
SPECIAL REBUILT PRICE **$115**

PAINT MIXING MOTOR



DEVILBISS BINKS
List Price $290
SPECIAL REBUILT PRICE **$95**

INGERSOLL-RAND



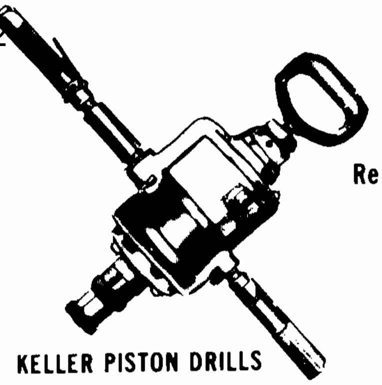
STARTING MOTOR
List Price $800
SPECIAL REBUILT PRICE **$345**

A R O

MODEL 7919
1/3 H.P. · 2500 R.P.M.
3/8'' · 24 Thd. Shaft
New $235.00 - REBUILT **$115**

AIR MOTORS

Rebuilt Guaranteed



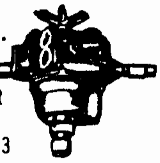
KELLER PISTON DRILLS

NON-REVERSIBLE
For Reversible Model
Add $50.00

Model	Cap.	RPM	Chuck or Morse Tapers	Weight	Rebuilt
OH	1/4	4500	3/8	5	$115
OS	1/4	2800	3/8	5	115
O	5/16	1800	3/8	5	115
OS1	3/8	900	1/2	6	115
OS2	7/16	625	1/2	6	115
OS3	5/8	375	5/8	6	120
1VH	1/2	1500	1/2	11	155
1V	1/2	1200	1/2	11	175
1VS1	9/16	750	5/8	12	195
1VS2	5/8	500	5/8	12	195
1VS3	7/8	350	5/8	12	200
2VH	29/32	1200	2M.T.	26	260
2V	29/32	700	2	26	260
2VS1	29/32	500	2	26	275
2VS2	1-1/4	400	3	28	275
2VS3	1-1/4	300	3	28	275
3VH	1	580	3	36	280
3V	1	490	3	37	280
3VS1	1-1/4	410	3	37	280
3VS2	1-1/2	330	4	38	305
3VS3	1-1/2	270	4	38	305
3VS4	2	150	4	40	335
3VS5	2-1/4	100	5	40	385

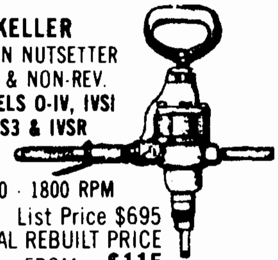
PISTON TOOLS

C.P.-THOR-I.R.



PISTON AIR MOTORS
#1, #2, #3
& #4 MT
Variable Speeds
SPECIAL REBUILT PRICE
FROM **$115**

KELLER

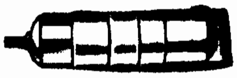


PISTON NUTSETTER
REV. & NON-REV.
MODELS O-IV, IVSI IVS3 & IVSR
350 · 1800 RPM
List Price $695
SPECIAL REBUILT PRICE
FROM **$115**

KELLER SCREWDRIVERS

PISTOL GRIP
NON-REVERSIBLE
Takes Standard Hex Bits
SPECIAL REBUILT PRICE **$24.95**

A R O



MODEL 7819
650 RPM
List Price $605
SPECIAL REBUILT PRICE **$75**

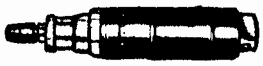
INGERSOLL-RAND

MODEL E9G
5.4 HP · 130 RPM
NON-REVERSIBLE
List Price $1995
SPECIAL REBUILT PRICE **$535**

CLECO

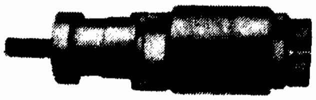
MODEL 401-NS
460 RPM
List Price $560
SPECIAL REBUILT PRICE **$115**

THOR



MODEL 5M
800 RPM
List Price $360
SPECIAL REBUILT PRICE **$195**

INGERSOLL-RAND



MODEL 2P
230 RPM
List Price $565
SPECIAL REBUILT PRICE **$155**

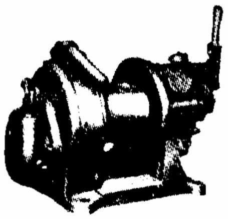
A R O

MODEL 7256 BS
1/3 H.P. · 750 R.P.M.
3/8'' · 24 Thd. Shaft
New $230 - REBUILT **$115**

REBUILT

AIR TUGGERS

GUARANTEED

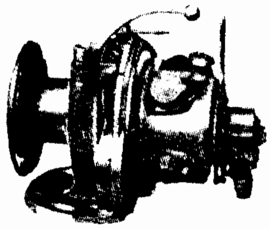


INGERSOLL — RAND TUGGER

LIST $2905

Size DU 750 lb Capacity Lifting Speed 40 Ft Per Min Weight 75 lbs. Cable Capacity 350 Ft of 3/16"

Special Rebuilt Price **$1350.00**

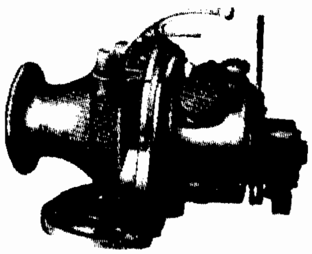


INGERSOLL — RAND TUGGER

LIST $4690

Size DUL 1000 lb Capacity Lifting Speed 75 Ft Per Min Weight 275 lbs Cable Capacity 290 Ft of 5/16"

Special Rebuilt Price **$1975.00**



INGERSOLL — RAND TUGGER

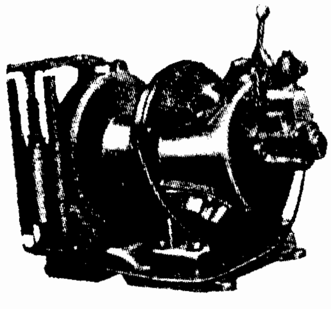
LIST $4145

Size DU 750 lb Capacity Lifting Speed 75 Ft Per Min Weight 270 lbs Cable Capacity 290 Ft of 5/16"

Rebuilt Price **$1475.00**

INGERSOLL — RAND TUGGER

LIST $6285

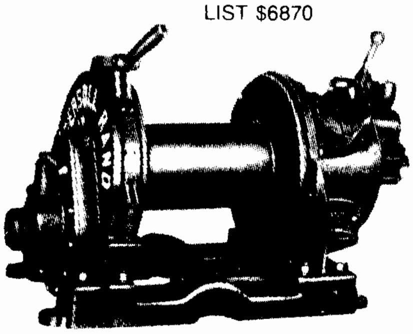


Size HU, 2000 lb Capacity
Lifting Speed 124 Ft Per Min
Weight 490 lbs
Cable Capacity 400 Ft of 7/16"

Special Rebuilt Price **$2995.00**

INGERSOLL — RAND TUGGER

LIST $6870

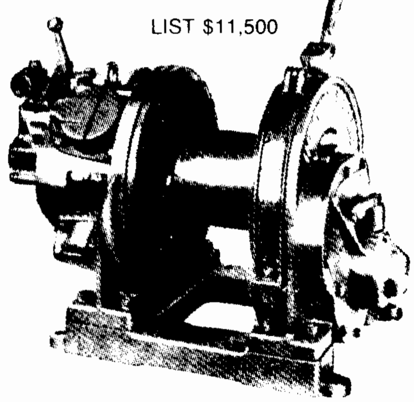


Size HUL, 2000 lb Capacity
Lifting Speed 124 Ft Per Min
Weight 562 lbs
Cable Capacity 850 Ft of 7/16"

Special Rebuilt Price **$3295.00**

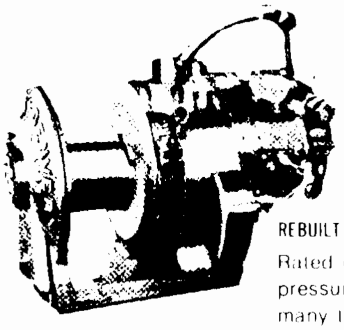
INGERSOLL — RAND TUGGER

LIST $11,500



Size K4U, 4000 lb. Capacity
Lifting Speed 95 Ft Per Min
Cable Capacity 550 Ft of 1/2"
Weight 850 lbs

Special Rebuilt Price **$5995.00**



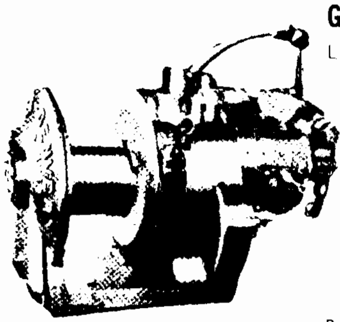
GARDNER DENVER

LIST $4600

Model HE Tugger and Utility Hoist for continuous heavy duty service

REBUILT **$2295.00**

Rated capacity 2500 lbs at 80 lbs air pressure Capacity can be increased many times by use of sheave or block and tackle



GARDNER DENVER

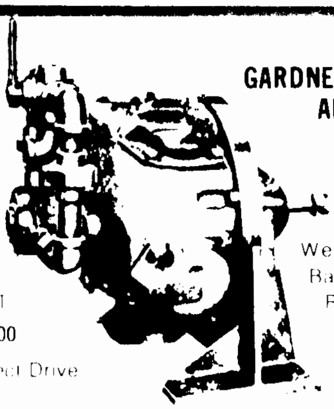
LIST $2425

Model HB

Rated capacity 1000 lbs at 80 lbs air pressure

Capacity can be increased many times by use of sheave or block tackle

Barrys Special Rebuilt Price **$1150.00**



GARDNER-DENVER AIR MOTOR

Reversible
Model MKB
2500 RPM
10 H P
Weight 180 lbs
Barrys Special Rebuilt Price
$715.00

LIST
$1900
Direct Drive

INGERSOLL-RAND AIR MOTOR

Size CM 2.9 H P
LIST
$1250

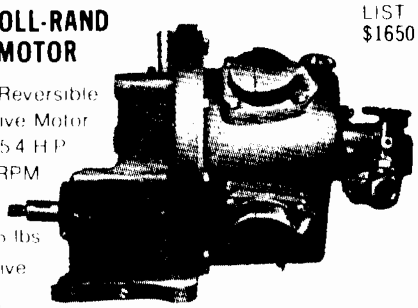
Direct Drive

1400 to 3200 RPM
Weight 130 lbs
Special Rebuilt Price **$650.00**

INGERSOLL-RAND AIR MOTOR

LIST
$1650

E9G Non-Reversible Geared Drive Motor
Size E9G 5.4 H P
130 - 285 RPM



Weight 205 lbs
Geared Drive

Barrys Special Rebuilt Price **$835.00**

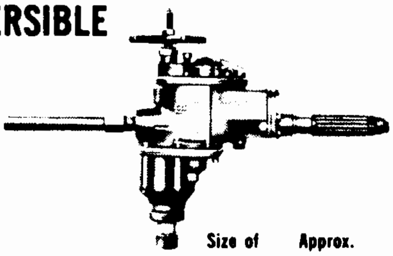
HEAVY DUTY DRILLS

REBUILT

1/2"TO 3"CAPACITY

Heavy Duty Air Drills are preferred for production work because they have the absolute smoothness of operation that is so necessary for accurate drilling, reaming, tapping, nut running, etc. At the same time, they possess the power to keep them up to speed when under a load. They are light in weight and easy to handle. Their power and efficiency make them ideal tools for many jobs that formerly required larger and heavier machines.

NON-REVERSIBLE

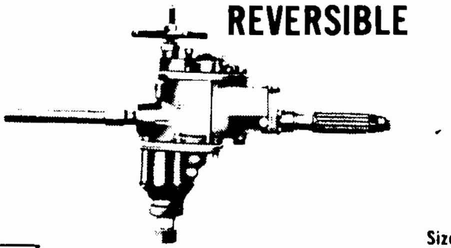


CAP.	Manufacturer	Model	RPM	Size of M.T.	Approx. List	Rebuilt
1/2"	CHICAGO PNEUMATIC	310-P	925	1 or Chuck	$2200.00	$ 495.00
	CHICAGO PNEUMATIC	310-S	925	1 or Chuck	2200.00	495.00
9/16"	CHICAGO PNEUMATIC	315	750	1 or Chuck	2460.00	650.00
	CHICAGO PNEUMATIC	323-S	550	1 or Chuck	2460.00	650.00
	CHICAGO PNEUMATIC	323-S	700	1 or Chuck	2460.00	650.00
	CHICAGO PNEUMATIC	315	500	1 or Chuck	2460.00	650.00
	INGERSOLL RAND	2XL	500	2	2370.00	650.00
	INGERSOLL RAND (New)	2XK	725	1	2370.00	775.00
	THOR	253G	700	Chuck	2370.00	625.00
	THOR	253X	700	1	2370.00	625.00
	THOR	353X	700	1	2370.00	675.00
	THOR	353Y	550	1	2370.00	675.00
7/8"	CHICAGO PNEUMATIC	323-S	350	1 or Chuck	2875.00	700.00
	CHICAGO PNEUMATIC	315	315	1 or Chuck	2875.00	725.00
	INGERSOLL RAND	2XM	350	2	2370.00	725.00
	THOR	353Z	350	1	2370.00	675.00
29/32"	CHICAGO PNEUMATIC	327	700	2	3750.00	950.00
	CHICAGO PNEUMATIC	326	700	2 or 3	3750.00	900.00
	INGERSOLL RAND	3H	800	2	3890.00	1075.00
	INGERSOLL RAND	3J	450	2	3890.00	1075.00
	THOR	362H2	700	2	3790.00	900.00
1"	CHICAGO PNEUMATIC	326	450	2	4160.00	1025.00
	CHICAGO PNEUMATIC	327	450	3	4160.00	1075.00
	INGERSOLL RAND	3SJ	450	3	4260.00	1075.00
	INGERSOLL RAND	3SH	800	3	4260.00	1075.00
	THOR	362X	450	3	4260.00	1025.00
1-1/4"	CHICAGO PNEUMATIC	327	375	3	4890.00	1175.00
	INGERSOLL RAND	4J	450	3	4900.00	1175.00
	INGERSOLL RAND	M4J	450	4	4900.00	1175.00
	INGERSOLL RAND	3SM	185	3	4200.00	1650.00
	INGERSOLL RAND	4K	310	3	4900.00	1275.00
	INGERSOLL RAND	3SK	300	3	3900.00	1175.00
	ROTOR	884	200	4	3490.00	925.00
	THOR	363Z	250	3	3690.00	1050.00
	THOR	363Y	350	3	3690.00	1075.00
	THOR	363X3	450	3	3690.00	1100.00
2"	CHICAGO PNEUMATIC	350	275	4	5175.00	1775.00
	CHICAGO PNEUMATIC	350	450	4	5175.00	1775.00
	INGERSOLL RAND	4SM	160	4	5150.00	1800.00
	INGERSOLL RAND	M5J	450	4	5150.00	1750.00
	THOR	385Y	300	4	4890.00	1375.00
	THOR	385Y3	300	3	4890.00	1375.00
2-1/2"	CHICAGO PNEUMATIC	350	150	4	5175.00	1800.00
	INGERSOLL RAND	5-SM	120	5	5175.00	1900.00

SPECIAL DRILLS AVAILABLE

OTHER MODELS AVAILABLE

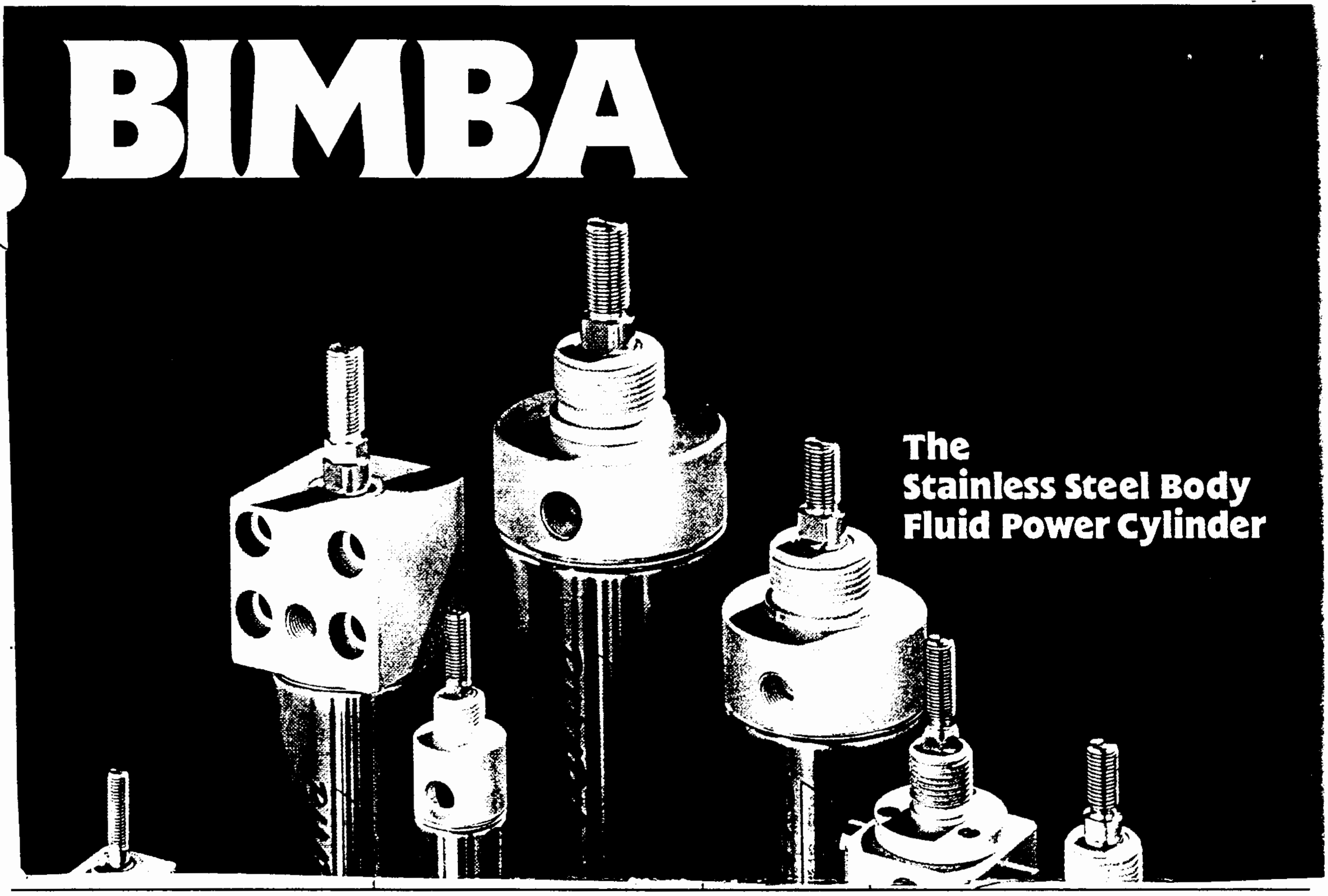
REVERSIBLE



CAP	Manufacturer	Model	RPM	Size of M.T.	Approx. List	Rebuilt
9/16"	CHICAGO PNEUMATIC	315R	750	2	$2350.00	$ 700.00
	INGERSOLL RAND	22H	1500	Chuck	2390.00	725.00
	INGERSOLL RAND	22K	725	1 or 2	2390.00	725.00
	ROTOR	40C	700	2	1965.00	585.00
	ROTOR	20	500	1	1965.00	585.00
	THOR	353RX	700	1 or 2	1965.00	675.00
7/8"	CHICAGO PNEUMATIC	315R	200	3	2415.00	875.00
	CHICAGO PNEUMATIC	315R	350	2	2415.00	875.00
	CLECO	101AR	300	2	1890.00	695.00
	INGERSOLL RAND	22L	550	2	2390.00	750.00
	THOR	353RZ	200	3	2360.00	725.00
15/16"	CHICAGO PNEUMATIC	315R	100	3	2515.00	895.00
29/32"	CHICAGO PNEUMATIC	327R	700	2 or 3	4160.00	1025.00
	CHICAGO PNEUMATIC	326R	650	3	4160.00	950.00
	INGERSOLL RAND	33J	450	2	4160.00	1250.00
	THOR	362RH2	700	2	4160.00	950.00
1"	CHICAGO PNEUMATIC	327R	500	3	4290.00	1150.00
	INGERSOLL RAND	33SH	800	3	4290.00	1150.00
	INGERSOLL RAND	33SJ	450	3	4290.00	1150.00
	THOR	362RX3	450	3	4190.00	1075.00
	THOR	263RM	450	3	4190.00	1050.00
	THOR	363RY3	350	3	4190.00	1175.00
1-1/4"	CHICAGO PNEUMATIC	327R	400	3	4390.00	1300.00
	CHICAGO PNEUMATIC	327R	290	3	4390.00	1300.00
	CHICAGO PNEUMATIC	327R	160	4	4555.00	1825.00
	CLECO	103AR	160	3	4555.00	1250.00
	INGERSOLL RAND	44SL	225	4	5300.00	1850.00
	INGERSOLL RAND	33SM	185	3 or 4	4345.00	1850.00
	INGERSOLL RAND	33M4	185	4	4210.00	1850.00
	INGERSOLL RAND	44J	450	3	5300.00	1275.00
	INGERSOLL RAND	33SK	300	3	4290.00	1275.00
2"	CHICAGO PNEUMATIC	350R	275	4	5475.00	1880.00
	CHICAGO PNEUMATIC	350R	450	4	5475.00	1880.00
	CHICAGO PNEUMATIC	350R	100	4	5475.00	1900.00
	INGERSOLL RAND	M55K	300	4	5775.00	1975.00
	INGERSOLL RAND	44SM	160	4	5450.00	1925.00
	THOR	385RY	310	4	5190.00	1425.00
	THOR	264RY	130	4	5190.00	1375.00
2-1/2"	CHICAGO PNEUMATIC	327R	55	4	5815.00	1975.00
	INGERSOLL RAND	55L	200	4	5775.00	2075.00
3"	CHICAGO PNEUMATIC	350R	100	5	5815.00	1675.00
	INGERSOLL RAND	55-SM	120	5	5775.00	1975.00
	THOR	385RL3	125	3	5410.00	1600.00
OVER 3"	CHICAGO PNEUMATIC	350R	65	5	5815.00	1900.00
	INGERSOLL RAND	55R	25	5	5775.00	2075.00
	THOR	385RL5	75	5	5775.00	1875.00

ALL PRICES QUOTED ARE F.O.B. OUR DETROIT PLANT

BARRY AIR TOOL SALES • 10650 CLOVERDALE • DETROIT, MICHIGAN 48204

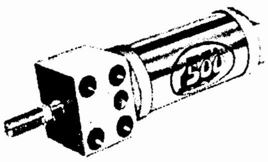
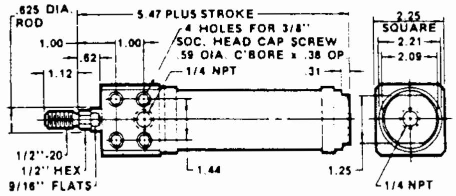
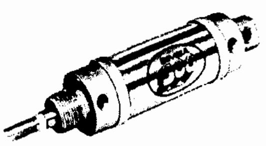
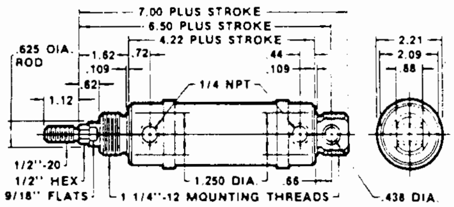
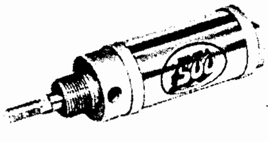
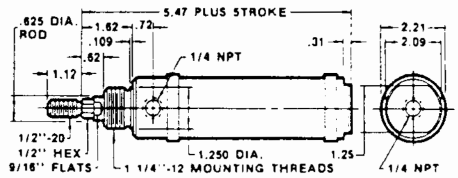


2" BORE

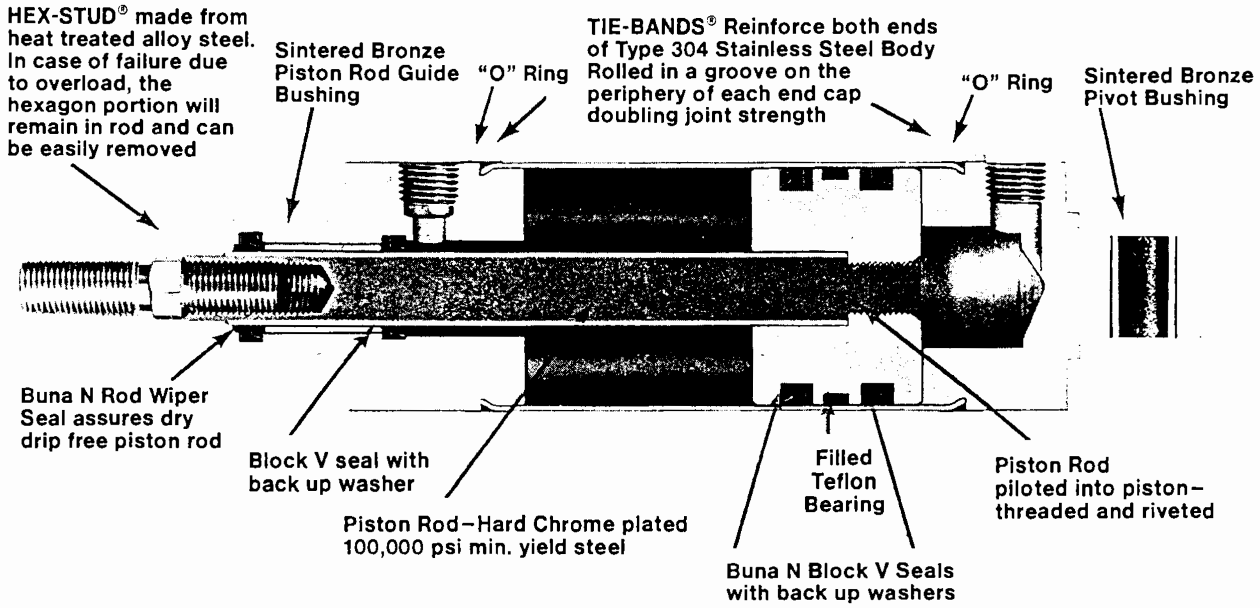
☐ Enter Stroke Length as 3rd Digit

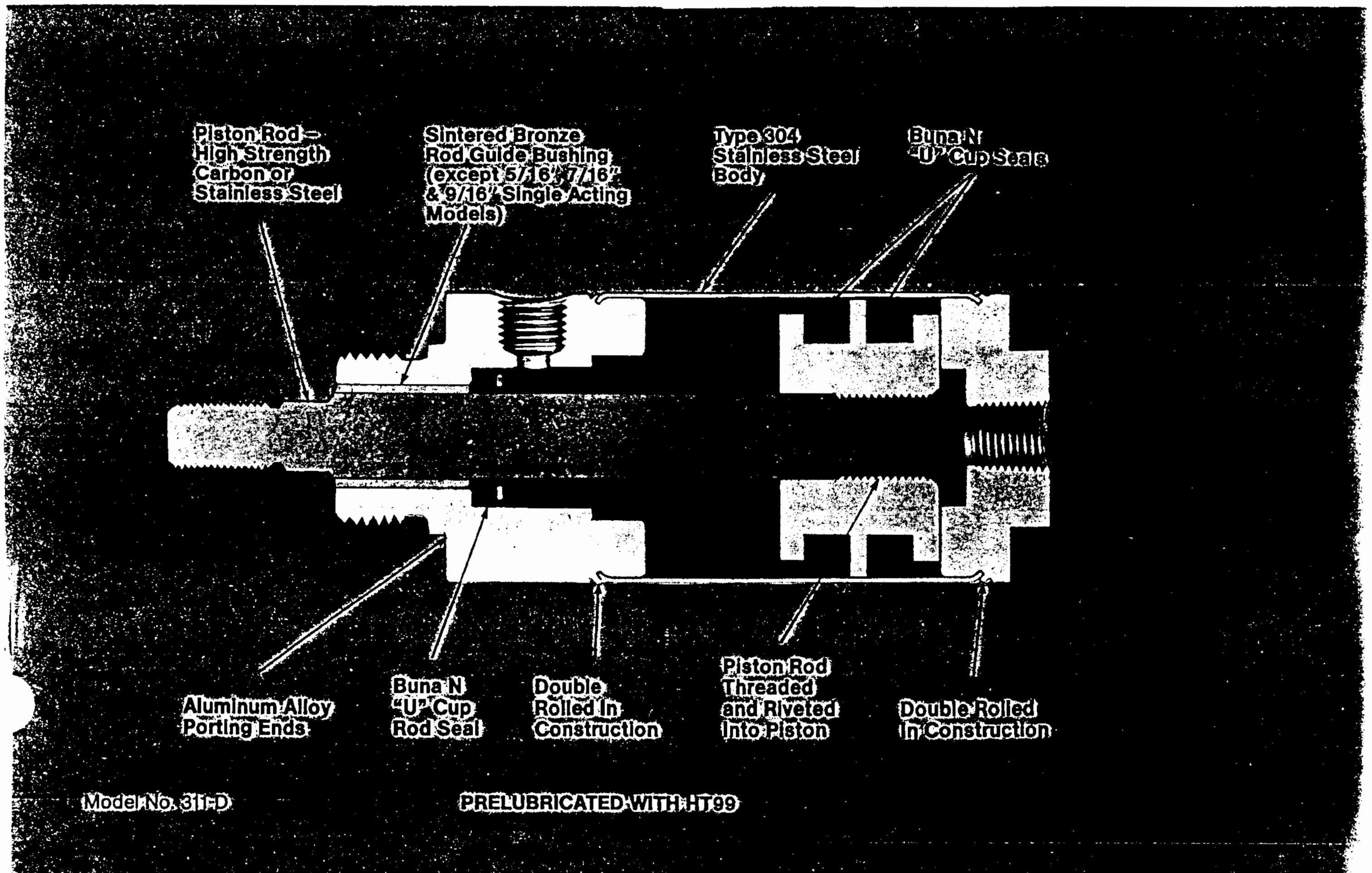
"500" HYDRAULIC CYLINDERS

- Push Force – 3.14 x psi, Pull Force – 2.83 x psi
- Rated 500 psi Hydraulic (non-shock), Double Acting

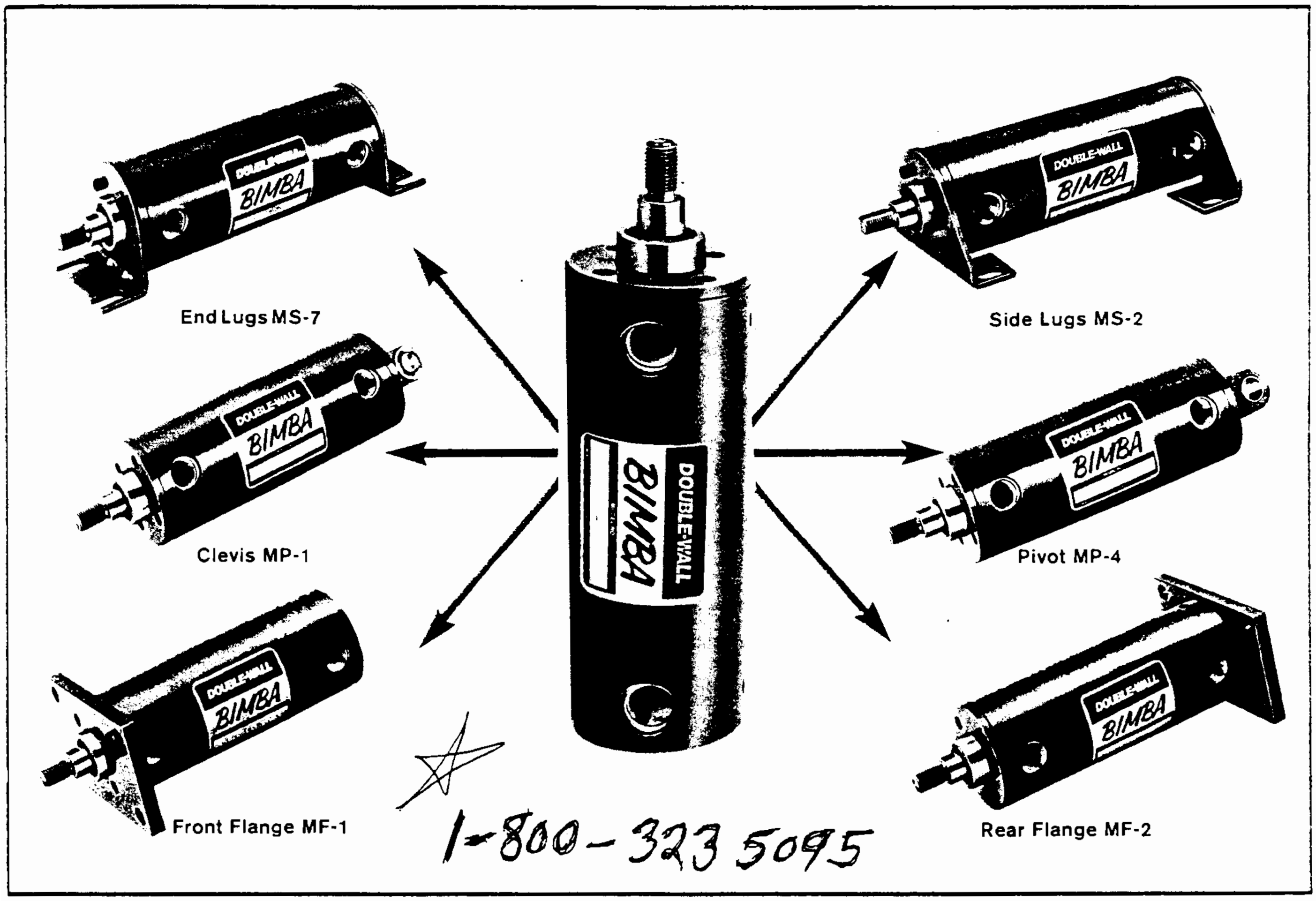
MODEL	DESCRIPTION	PRICE	DIMENSIONS
H-31☐-DBZ 	**Front Block Mount** Standard Stroke Lengths: 1" increments to 6" Maximum Stroke 12"	**$84.25** BASE PRICE Add $2.30 per inch of stroke	
H-31☐-DUZ 	**Universal Mount**–Double End or Rear Pivot– Standard Stroke Lengths: 1" increments to 12" Maximum Stroke 32" Optional Accessories: D-8325-A Pivot Bracket D-8319 Mounting Bracket D-8313-A Clevis	**$78.90** BASE PRICE Add $2.30 per inch of stroke	
H-31☐-DZ 	**Nose Mount** Standard Stroke Lengths: 1" increments to 6" Maximum Stroke 12" Optional Accessory: D-8319 Mounting Bracket	**$73.65** BASE PRICE Add $2.30 per inch of stroke	

500® HYDRAULIC CYLINDERS

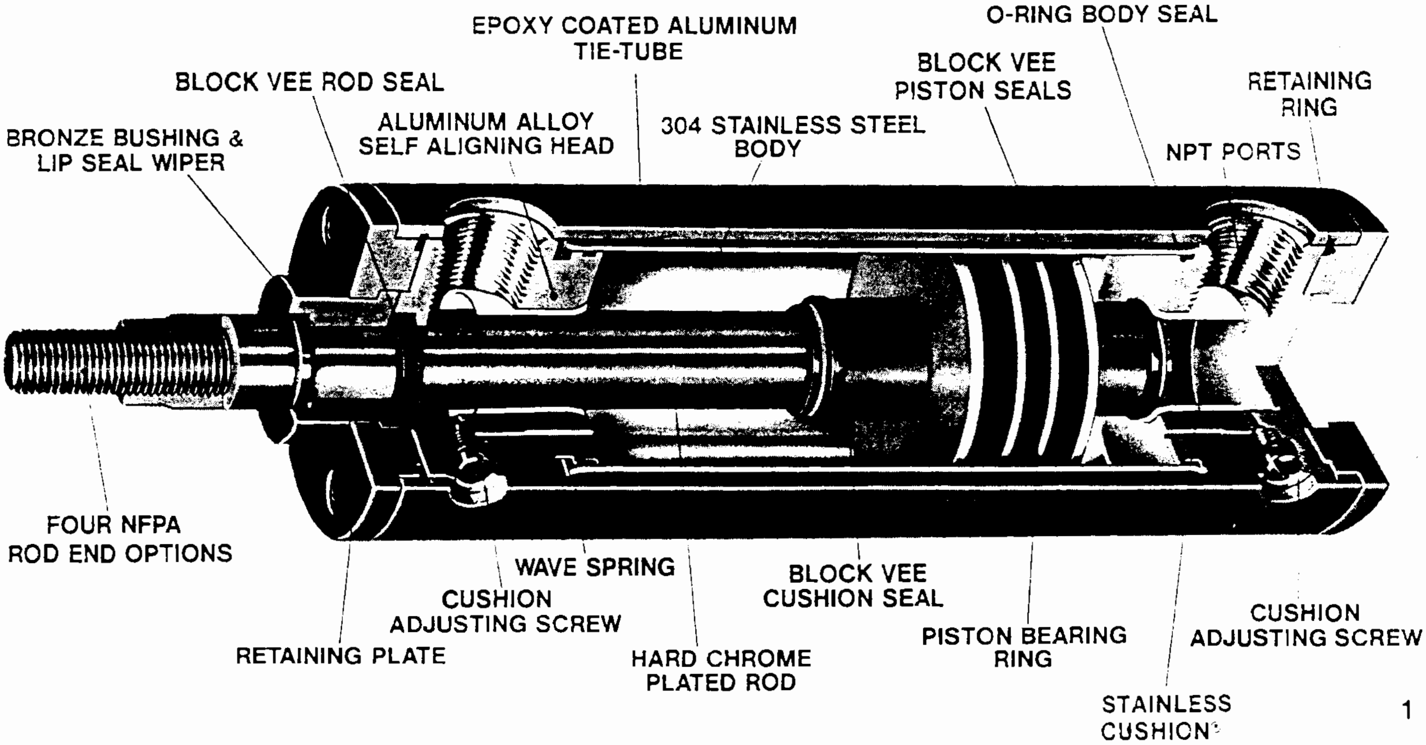




REPLACE THOSE BULKY TIE-ROD CYLINDERS WITH THE STREAMLINED BIMBA DOUBLE-WALL® TIE-TUBE DESIGN AIR CYLINDER. ONE BASIC CYLINDER CONVERTS INTO SIX NFPA MOUNTING STYLES.



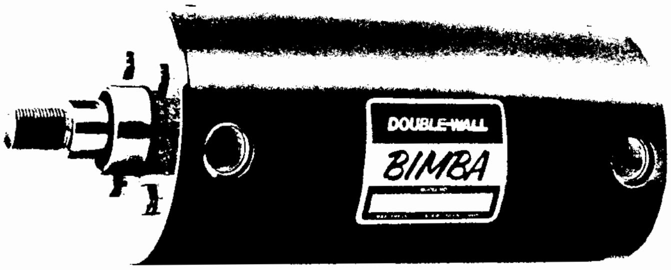
ANATOMY OF THE BIMBA DOUBLE-WALL®



4" BORE DOUBLE-WALL® SERIES DW-125

- TIE-TUBE - Epoxy Coated Aluminum
- BODY - Smooth I.D. Type 304 Stainless Steel
- PRESSURE RATING - 200 PSI Air
- POWER FACTOR - 12.5 of Air Line Pressure
- PISTON ROD - Hard Chrome Plated, Standard
- BUNA N SEALS - Viton see Page 9 for Prices
- CUSHIONS — Exclusive Bimba STAINLESS CUSHION®. Available Either or Both Ends, see Page 2 "How to Order" and Page 9 for Prices
- STROKE LENGTHS - 1" Increments Thru 24" Standard, Long and Fractional Strokes Available on Request
- ROD END OPTIONS - Four NFPA Rod Ends Available, see Page 2 for Price and Dimensions. See Important Information, Inside Back Cover for Special Rod and Rod End Availability. One-Piece Threaded Male Rod (style #2) Shown Below is Shipped Unless Otherwise Specified.
- MOUNTING KITS - Six NFPA Mountings Available in Kits (including necessary hardware) for Attachment to Basic Cylinder. MOUNTING KITS AND BASIC CYLINDERS ARE ORDERED AND SHIPPED AS SEPARATE ITEMS.

BASIC DOUBLE-WALL® CYLINDER

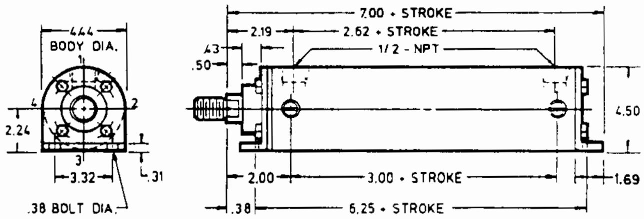


BASE PRICE $134.00 PLUS $4.55 per inch of stroke

(See Page 2 - How To Order)

MODEL	DESCRIPTION	PAGE
4" BORE MOUNTING KITS		
MSL-125	Side Lug Mount	7
MEL-125	End Lug Mount	7

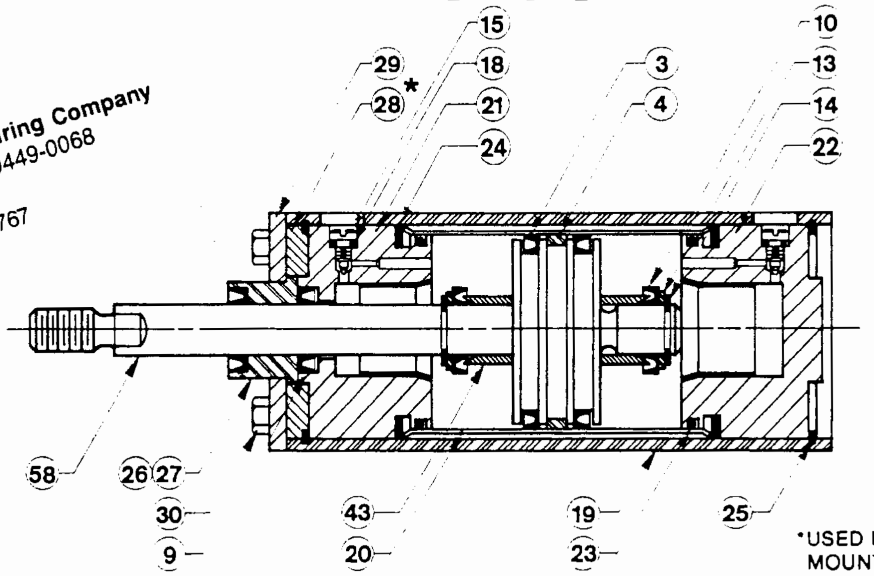
END LUG MOUNTS (NFPA MS-7)



MOUNTING KIT #MEL-125 PRICE $20.35

BASIC DOUBLE-WALL® CYLINDER DRAWINGS

DW SERIES



Bimba Manufacturing Company
Monee, Illinois 60449-0068
708/534-8544
FAX: 708/534-5767

Bimba Limited
23 Maxwell Road
Woodston, Peterborough
Cambridgeshire PE2 7JD
United Kingdom
0733 391078
FAX: 0733 391080

*USED IN MOST MOUNTING KITS AND ON ALL 3¼" AND 4" BORE BASIC CYLINDERS.

from a fluid power manual by Anthony Esposito

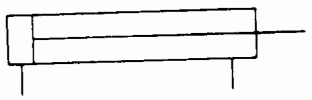
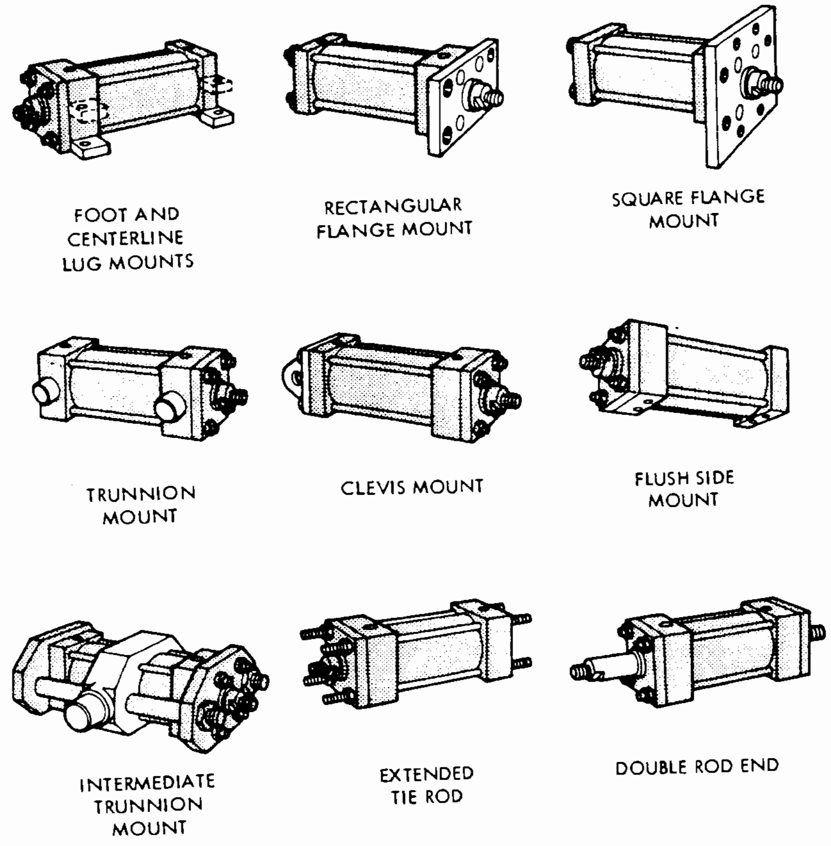


Fig. 7-3. Symbolic representation of double-acting cylinder. (Courtesy of Sheffer Corporation, Cincinnati, Ohio.)

smooth deceleration at both ends of the stroke. Therefore, the piston does not bang into the end caps with excessive impact, which could damage the hydraulic cylinder after a given number of cycles. The symbol for a double-acting cylinder is shown in Fig. 7-3. Notice that the symbol implies how the cylinder operates without showing any details. In drawing hydraulic circuits (as done in Chapter 9), symbolic representation of all components will be used. This facilitates circuit analysis and troubleshooting. Also, it would be too time-consuming to draw each component schematically. The symbols, which are merely combinations of simple geometric figures such as circles, rectangles, and lines, make no attempt to show the internal configuration of a component. However, symbols must clearly show the function of each component.

Various types of cylinder mountings are in existence, as illustrated in Fig. 7-4. This permits versatility in the anchoring of cylinders. The rod ends are usually

Fig. 7-4. Various cylinder mountings. (Courtesy of Sperry Vickers, a Division of Sperry Rand Corp., Troy, Michigan.)



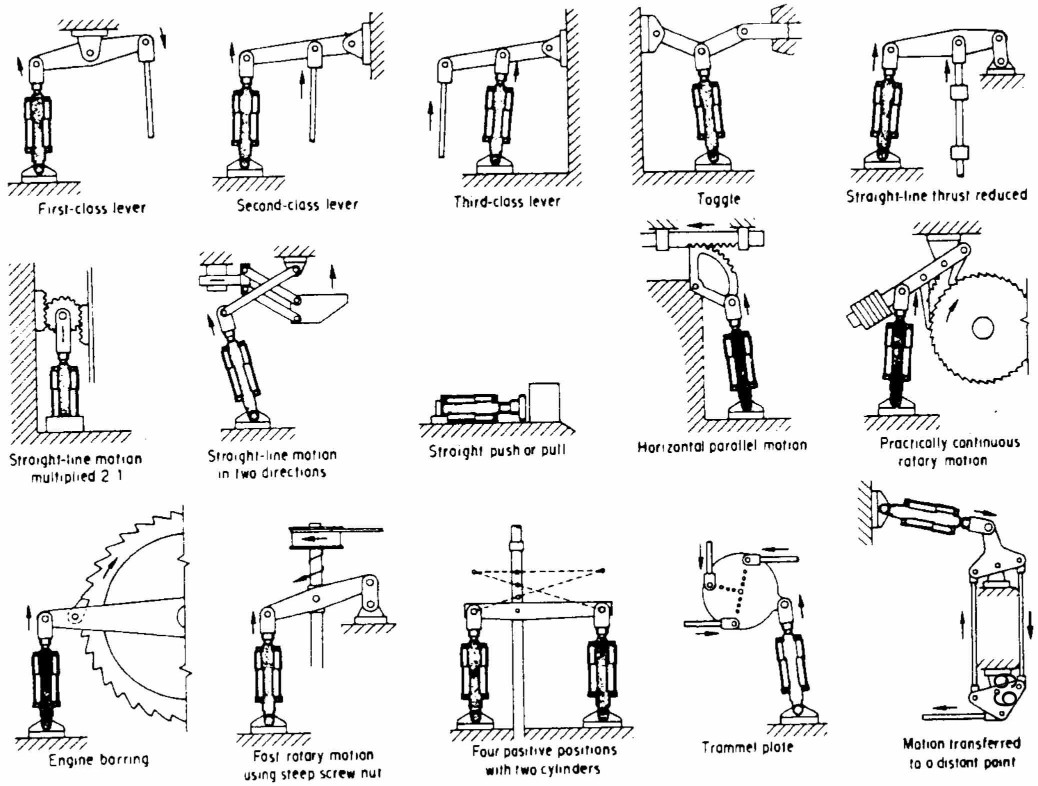


Fig. 7-5. Typical mechanical linkages that can be combined with hydraulic cylinders. (Courtesy of Rexnord Inc., Hydraulic Components Division, Racine, Wisconsin.)

threaded so that they can be attached directly to the load, a clevis, a yoke, or some other mating device.

Through the use of various mechanical linkages, the applications of hydraulic cylinders are limited only by the ingenuity of the fluid power designer. As illustrated in Fig. 7-5, these linkages can transform a linear motion into either an oscillating or rotary motion. In addition, linkages can also be employed to increase or decrease the effective leverage and stroke of a cylinder.

Much effort has been made by manufacturers of hydraulic cylinders to reduce or eliminate the side loading of cylinders created as a result of misalignment. It is almost impossible to achieve perfect alignment even though the alignment of a hydraulic cylinder has a direct bearing on its life.

A universal alignment mounting accessory designed to reduce misalignment problems is illustrated in Fig. 7-6. By using one of these accessory components and a

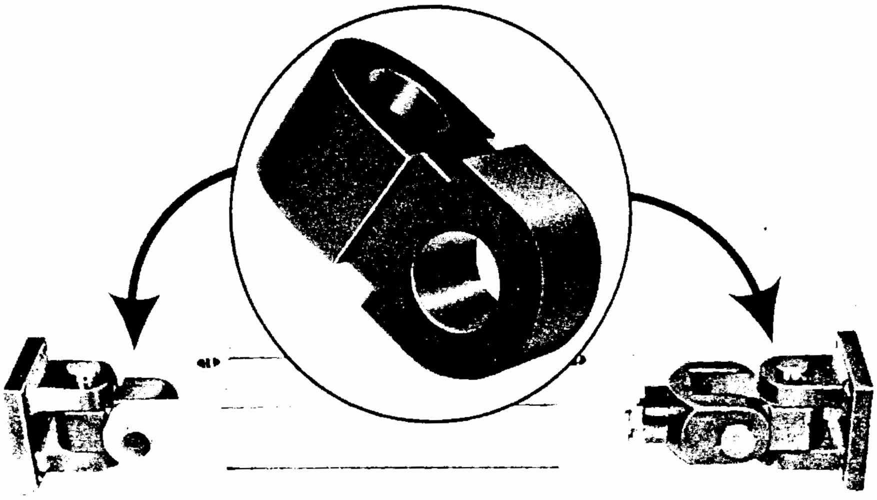


Fig. 7-6. Universal alignment mounting accessory for fluid cylinders. (Courtesy of Sheffer Corporation, Cincinnati, Ohio.)

mating clevis at each end of the cylinder (see Fig. 7-6), the following benefits are obtained:

1. Freer range of mounting positions
2. Reduced cylinder binding and side loading
3. Allowance for universal swivel
4. Reduced bearing and tube wear
5. Elimination of piston blow-by caused by misalignment

The force output and piston velocity of double-acting cylinders is not the same for extension and retraction strokes. This phenomenon is due to the effect of the rod and is defined by Eqs. (7-1)–(7-4).

Extension stroke:

$$\text{force (lb)} = \text{pressure(psi)} \times \text{piston area (in.}^2\text{)} \tag{7-1}$$

$$\text{velocity (ft/sec)} = \frac{\text{input flow (ft}^3\text{/sec)}}{\text{piston area (ft}^2\text{)}} \tag{7-2}$$

Retraction stroke:

$$\text{force (lb)} = \text{pressure (psi)} \times [\text{piston area (in.}^2\text{)} - \text{rod area (in.}^2\text{)}] \tag{7-3}$$

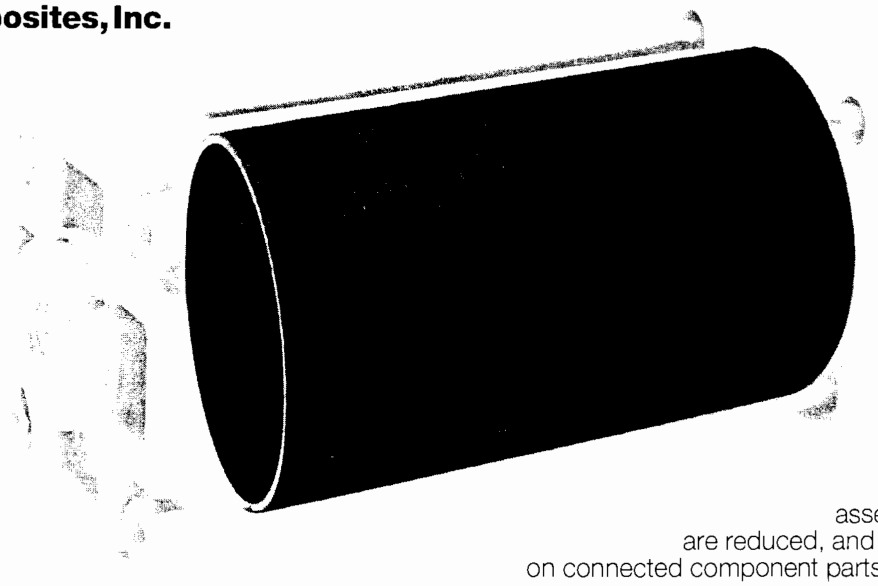
$$\text{velocity (ft/sec)} = \frac{\text{input flow (ft}^3\text{/sec)}}{\text{piston area (ft}^2\text{)} - \text{rod area (ft}^2\text{)}} \tag{7-4}$$

Black-Amalgon™

HYDRAULIC AND PNEUMATIC CLYLINDER TUBING

The Better Choice...

from Amalga Composites, Inc.



Black-Amalgon™, the Alternative to Metal

For more than 20 years, there has been an alternative to metallic pneumatic and hydraulic cylinder tubing for low-, medium- or high-pressure systems: Black-Amalgon™.

Constructed of fiber-reinforced thermoset epoxy matrix, Black-Amalgon has an inner layer of evenly dispersed low-friction additives. The result: A light-weight, high-strength, corrosion-resistant composite material which can easily replace carbon steel, honed and chromed steel, stainless steel, aluminum, or brass cylinder barrels.

Black-Amalgon for low pressure applications is constructed of a fiberglass/epoxy composite. Depending upon the pressure requirements, medium pressure and high pressure versions may incorporate other materials in hybrid configurations.

Four Big Reasons

1. **75% Reduction in Weight** — Right off the bat, Black-Amalgon saves you money in material costs, handling, shipping and storage. At approximately 1/4 the weight of steel or brass, and 3/4 the weight of aluminum, Black-Amalgon is much easier to handle than traditional metal tubing. Therefore, freight costs are lower, assembly times are reduced, and stress loads on connected component parts decrease.
2. **Superior Corrosion Resistance** — You can expect trouble-free performance in chemical, high moisture and other adverse environments that would normally corrode or impair the operation of metals. No problem with salt or chlorinated water either. With Black-Amalgon, you'll see significant reduction in life-cycle costs.
3. **Reduced Maintenance Costs** — Say "good bye" to piston lock-up problems. Black-Amalgon's patented manufacturing process ensures a smooth, self-lubricating, homogeneous internal surface that prevents pistons from sticking, even after they've remained idle for some time.
4. **Eliminate Honing Costs** — A surface smoother than honed steel...without the cost of honing. A 5-15 RMS micro-inch ID finish performs just like a honed or chromed surface.

Chemical Resistance

Available Cylinder Tubing Material

CHEMICALS	BRASS	ALUMINUM	EPOXY
Nitric Acid 10%	**NR**	**NR**	**R**
Phosphoric Acid 20%	**NR**	**NR**	**R**
Acetic Acid 10%	**NR**	R	**R**
Sodium Hydroxide	Fair	**NR**	**R**
Amonium Hydroxide	**NR**	**NR**	**R**
Calcium Sulfate	R	R	**R**
Sodium Nitrate	R	R	**R**
Sodium Chloride	**NR**	**NR**	**R**
Ammonium Chloride	**NR**	**NR**	**R**
Calcium Hypochloride	**NR**	**NR**	**R**
Water	R	R	**R**
Gasoline	R	R	**R**
Motor Oil	R	R	**R**
Hydraulic Fluid	R	R	**R**

R — Recommended

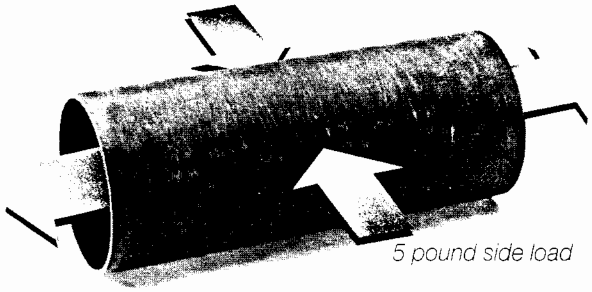
NR —Non-recommended

The list at right is for normal temperatures and operating conditions. For Black-Amalgon's resistance to other chemicals and various temperatures, consult the company for recommendations.

Need More Reasons to Choose Black-Amalgon?

- **No More Catastrophic Failures** — When metal tubing catastrophically fails dangerous shrapnel may result. Should Black-Amalgon fail, there is micro-cracking which simply releases pressure to safe levels.
- **Shape Stability and Impact Resistance** — Ship, store and cut Black-Amalgon — it will retain its circular shape. And unlike metals, the product simply does not dent. Material impact strength is 40 izod ft. lbs.
- **Easy Bonding; Uses Standard End-Fittings** — The material can be easily bonded to most metals using a readily available two-part epoxy bonding kit. (Contact us to discuss bonding to non-metallic materials.) Tie rod or bonded end-cap configurations are the common choice when Black-Amalgon is used as cylinder tubing.
- **Can be Pigmented or Painted** — Need a different color? No problem. Black-Amalgon can be pigmented or painted any color you need. Standard urethane-based paint works well for most applications.
- **Excellent Thermal Stability; Non-Interference** — With a very low coefficient of thermal expansion, Black-Amalgon operates efficiently from -100° F to +270° F. Customers have reported success in using the material in temperatures below -200° F. Unless requested otherwise, the material will not interfere with electronic or magnetic componentry.
- **Machinability** — Black-Amalgon can be cut, chamfered, ground to tight tolerances, shouldered, bored and threaded...to meet your engineering requirements.
- **Best Composite Process** — Black-Amalgon offers the best combination of strength-to-weight ratio, burst strength, and axial loading capability when compared to tubular structures made by other composite processes.

Black-Amalgon
Cycle Tested For Maximum Performance



10,000,000 cycles dry cylinder test without measurable cylinder wear.

Relative Thermal Conductivity of Samples of Black Amalgon In Comparison With Pure Metal Tubes

Material	Density ρ, lbs/in³	Thermal Conductivity K, BTU/ft•hr•°F
Amalgon	.072	0.250
Brass	.320	61.000
Steel	.280	30.000
Aluminum	.100	132.000
Zinc	.250	65.000
Copper	.320	223.000

Large Inventory of Tooling, Fast Deliveries

Over the years, we have established an extensive inventory of tooling to meet most needs. If not available, we will quote the cost of creating the tooling separately. Currently, tooling is in-house to manufacture Black-Amalgon from 1/2-inch ID to 30-inch ID. Metric sizes are also available. Wall thickness can be varied to meet pressure requirements or component geometry.

If you're in a hurry, lead times are often significantly less than required for metal structures because the self-lubricating, honed-like ID is achieved without lengthy honing. We can also stock products to meet your JIT requirements.

We Go To Any Lengths

Black-Amalgon is sold in random lengths or cut pieces. We will go to any length to provide superior product quality and responsive service.

Founded in 1966, Amalga Composites, Inc. has grown to be leader in designing, engineering and manufacturing filament-wound composites to meet the tough environmental and performance requirements of the fluid power industry.

Interested?

Call for additional technical supplements, sizing and pricing for Black-Amalgon and other composite products available from Amalga Composites, Inc. We're interested in helping you. Call 1-800-262-5424

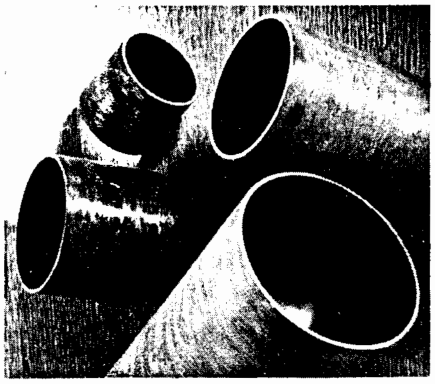
Innovative Composite Structures since 1966

AMALGA COMPOSITES, INC.
10600 West Mitchell Street, West Allis, WI 53214 Phone 414 453-9555 Fax 414 453-9561

PRODUCT SELECTION GUIDE

Black-Amalgon®

Amalga Composites, Inc. ... The Better Choice. ***BA.05-93***



lack-Amalgon® is a patented fiber-reinforced thermoset composite material specifically designed for use in cylinder tubing applications. Its material strengths, corrosion resistance, minimum 75% weight savings when compared to steel, and 5-15 RMS microinch bore finish offer significant benefits when compared to traditional metals.

OEM cylinder, actuator and motion control companies, OEM machine builders, end-user systems and equipment manufacturers have used Black-Amalgon® to replace steel, aluminum and brass for years.

More detailed information on strengths and material properties, corrosion resistance, and possible design modifications will be made promptly available upon your request.

STANDARD MEASUREMENT SYSTEM

Model No.[1]	Standard Bore Size (Inches)	Nominal Bore Size (Inches)	Bore Tolerances Available by Selected Procedures (Inches)	Nominal O.D. Size[2] (Inches)	Random Lengths To	Operating PSI Non-Tie Rod Design[3]	Operating PSI Tie Rod Design[3]	Weight Per Foot LBS	Model No.
BA100	1.000	1.000 1.003	-.000 +.002	1.250 1.253	5 feet	1220	2,440	.33	BA100
BA112	1.125	1.125 1.128	-.000 +.002	1.375 1.378	5 feet	1080	2,170	.37	BA112
BA125	1.250	1.250 1.256	-.000 +.002	1.500 1.506	5 feet	980	1,950	.41	BA125
BA150	1.500	1.500 1.506	-.000 +.002	1.750 1.754	5 feet	810	1,620	.50	BA150
BA175	1.750	1.750 1.756	-.000 +.003	2.000 2.004	5 feet	700	1,390	.56	BA175
BA200	2.000	2.000 2.006	-.000 +.003	2.250 2.254	10 feet	610	1,200	.66	BA200
BA225	2.250	2.250 2.258	-.000 +.003	2.500 2.506	10 feet	543	1,000	.75	BA225
BA250	2.500	2.500 2.508	-.000 +.003	2.750 2.760	10 feet	490	970	.83	BA250
BA275	2.750	2.750 2.758	-.000 +.004	3.000 3.010	10 feet	440	890	.91	BA275
BA300	3.000	3.000 3.008	-.000 +.004	3.250 3.260	10 feet	410	810	1.0	BA300
BA325	3.250	3.250 3.258	-.000 +.004	3.500 3.510	10 feet	380	750	1.1	BA325
BA350	3.500	3.500 3.508	-.000 +.005	3.750 3.760	10 feet	350	700	1.2	BA350
BA375	3.750	3.750 3.758	-.000 +.005	4.000 4.010	10 feet	330	650	1.3	BA375
BA400	4.000	4.000 4.010	-.000 +.005	4.250 4.260	10 feet	300	610	1.3	BA400
BA500	5.000	5.000 5.010	-.000 +.005	5.250 5.260	10 feet	240	490	1.7	BA500
BA575	5.750	5.750 5.760	-.000 +.005	6.000 6.012	10 feet	210	420	1.9	BA575
BA600	6.000	6.000 6.010	-.000 +.005	6.250 6.262	10 feet	200	410	2.0	BA600
BA700	7.000	7.000 7.012	-.000 +.007	7.250 7.262	10 feet	170	350	2.3	BA700
BA800	8.000	8.000 8.015	-.000 +.007	8.250 8.265	10 feet	150	300	2.7	BA800
BA1000-A	10.000	10.000 10.020	-.000 +.008	10.370 10.410	10 feet	190	330	5.0	BA1000-A
BA1000-B	10.000	10.000 10.020	-.000 +.008	10.490 10.530	10 feet	260	430	6.7	BA1000-B
BA1200-A	12.000	12.000 12.020	-.000 +.010	12.370 12.410	10 feet	160	280	6.0	BA1200-A
BA1200-B	12.000	12.000 12.020	-.000 +.010	12.490 12.530	10 feet	220	360	8.0	BA1200-B
BA1400-A	14.000	14.000 14.020	-.000 +.010	14.370 14.410	10 feet	140	240	7.0	BA1400-A
BA1400-B	14.000	14.000 14.020	-.000 +.010	14.490 14.530	10 feet	190	310	9.3	BA1400-B
BA1600-B	16.000	16.000 16.025	-.000 +.010	16.490 16.550	10 feet	170	270	10.7	BA1600-B
BA1600-C	16.000	16.000 16.025	-.000 +.010	16.620 16.690	10 feet	210	320	13.4	BA1600-C
BA1800-B	18.000	18.000 18.025	-.000 +.010	18.490 18.550	10 feet	150	240	11.5	BA1800-B

Model No.[1]	Standard Bore Size (Inches)	Nominal Bore Size (Inches)	Bore Tolerances Available by Selected Procedures (Inches)	Nominal O.D. Size[2] (Inches)	Random Lengths To	Operating PSI Non-Tie Rod Design[3]	Operating PSI Tie Rod Design[3]	Weight Per Foot LBS	Model No.
BA1800-C	18.000	18.000 18.025	−.000 +.010	18.620 18.690	10 feet	190	290	15.0	BA1800-C
BA2000-B	20.000	20.000 20.025	−.000 +.010	20.490 20.560	10 feet	130	220	13.5	BA2000-B
BA2000-C	20.000	20.000 20.025	−.000 +.010	20.620 20.700	10 feet	170	260	17.0	BA2000-C
BA2400-C	24.000	24.000 24.025	.000 .020	24.620 24.700	104 inches	140	220	19.1	BA2400-C
BA3000	30.000	30.000 30.025	.000 .020	31.000 31.080	104 inches	190	270	38.6	BA3000

METRIC MEASUREMENT SYSTEM

Model No.[1]	Standard Bore Size (Inches)	Nominal Bore Size (Inches)	Bore Tolerances Available by Selected Procedures (Inches)	Nominal O.D. Size[2] (MM)	Random Lengths To	Operating PSI Non-Tie Rod Design[3]	Operating PSI Tie Rod Design[3]	Weight Per Foot LBS	Model No.
MBA32	32	1.260 1.266	−.000 +.002	38.4	5 Ft.	960	1930	.44	MBA32
MBA40	40	1.575 1.581	−.000 +.002	46.4	5 Ft.	770	1550	.54	MBA40
MBA50	50	1.969 1.975	−.000 +.003	56.4	10 Ft.	610	1240	.65	MBA50
MBA63	63	2.480 2.488	−.000 +.003	69.4	10 Ft.	490	980	.82	MBA63
MBA72	72	2.835 2.843	−.000 +.004	78.4	10 Ft.	454	845	.71	MBA72
MBA80	80	3.150 3.158	−.000 +.004	96.4	10 Ft.	380	770	1.1	MBA80
MBA100	100	3.937 3.947	−.000 +.005	106.4	10 Ft.	310	620	1.3	MBA100
MBA125	125	4.921 4.931	−.000 +.005	121.4	10 Ft.	250	500	1.6	MBA125
MBA125-A	125	4.921 4.931	−.000 +.005	134.5	10 Ft.	400	690	2.4	MBA125-A
MBA160	160	6.299 6.311	−.000 +.005	166.4	10 Ft.	190	390	2.0	MBA160
MBA160-A	160	6.299 6.311	−.000 +.005	169.5	10 Ft.	310	540	3.1	MBA160-A
MBA160-B	160	6.299 6.311	−.000 +.005	172.7	10 Ft.	430	680	4.2	MBA160-B
MBA200	200	7.874 7.889	−.000 +.008	206.4	10 Ft.	150	310	2.5	MBA200
MBA200-A	200	7.874 7.889	−.000 +.008	209.5	10 Ft.	250	430	3.8	MBA200-A
MBA200-B	200	7.874 7.889	−.000 +.008	212.7	10 Ft.	340	550	5.1	MBA200-B

FOOTNOTES: 1. Under 1.000-inch tooling available. Tooling constantly upgraded. Call for availability. 2. Wall thickness can be changed to meet pressure or geometry requirements, from minimum .020 (.5mm) depending upon ID. 3. Operating pressures calculated with minimum 4:1 safety factor.

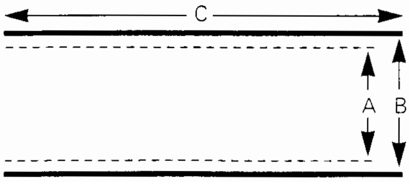
LET US QUOTE YOUR APPLICATION!

FAX this to 414-453-9561.

NAME: ______________________

CO.: ______________________

PHONE: ______________________



DIMENSIONS

A __________ B __________ C __________

CHAMFER: YES NO ______ DEGREES

SHOULDER: YES NO ______ DEPTH LENGTH __________

HOLES: YES NO ______ DIAMETER LOCATION __________

OTHER OPERATIONS: ______________________ QUANTITY IN PRODUCTION: __________

TOLERANCE REQUIREMENTS: O.D. ______ I.D. ______ LENGTH __________ OTHER ______________

SAFETY FACTOR: ___ : ___ CYLINDER APPLICATION? YES NO

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Air-Driven Actuators

The oldest type of air-driven actuator is probably a piston sliding in a cylinder, similar to actuators used for hydraulic systems. For pneumatic operation this type of actuator requires a seal or packing at the piston to keep the leakage at this point down to a minimum. This, of course requires good lubrication. The lubrication necessary for good sealing also reduces friction. Corrosion may be a serious problem for pneumatic actuators and should always be considered when selecting material for these actuators.

Double-Acting Diaphragm Cylinders

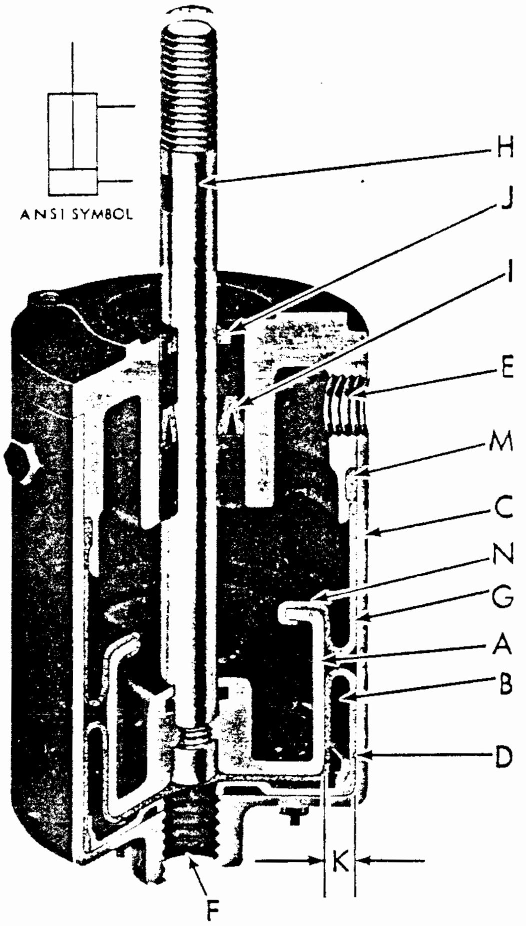


Fig. 5–15. Sectional view of air-operated double-acting diaphragm cylinder. (Courtesy of Bellofram Corporation)

Fig. 5–15 is a cutaway view of an air-operated, double-acting diaphragm cylinder. This type of actuator is virtually frictionless and may be designed in many configurations and sizes for different applications. Power loss due to friction is almost negligible, consisting only of the small amount of friction at the piston rod and the minute amount of power required to flex the tough, resilient diaphragm material.

The stroke is long in proportion to the rolling diaphragm height. For instance, a 4-inch (or 10 cm) diameter and 4-inch (or 10 cm) high diaphragm will provide a stroke of about 7 inches (or 18 cm). Similarly, a 6-inch (or 15 cm) diameter and 6 inch (or 15 cm) high diaphragm will provide a total stroke of 10-3/16 inches (or 26 cm). The manufacturers claim a working temperature range from −65°F (or −65°C) to 550°F (or 288°C), and in some cases a range of −120°F (or −85°C) to 700°F (or 370°C) is possible.

There are carefully designed diaphragm materials to meet a great variation in applied pressure, ranging from as little as about 2 torr to 500 psi (35 kg/cm²). Because the major portion of the dia-

phragm carrying the working load is supported by the piston area, the pressure may in some cases be as high as 1,200 psi (85 kg/cm^2). Service life is also high, in some cases up to 100 million cycles.

Operation of the cylinder shown in Fig. 5–15 is very simple. Assume that line air pressure is supplied to port *F*. Diaphragm *D* will then roll off piston *A* onto cylinder wall *C*. Diaphragm *G* will roll off cylinder wall *C* onto piston *A*.

According to Pascal's Law, air in a confined space is distributed equally in all directions. Therefore the pressure in space B keeps the piston in a central position within the cylinder without need for bearings to guide it. The only friction, therefore, is the force required to flex the diaphragms plus the small amount of friction between the resilient rod seal and bushing and piston rod *H* as it moves up.

To reverse this action, port *E* previously used as the exhaust port would receive line air pressure and become the inlet port. Port *F*, previously the inlet port, would then be the exhaust port. This reversal is easily done with a four-way valve operated either manually or automatically depending on the application.

With proper valving and controls this type of linear actuator can be made to reciprocate at selected rates of cycling, but in the majority of applications such as positioning, applying pressure, holding and retracting, the time interval between forward and reverse strokes is fairly long and subject to variations. In such cases the actuation is often manually controlled.

As shown in Fig. 5–12, this is a double-acting cylinder where pneumatic pressure moves the piston in both directions, forward and reverse. Therefore the piston rod *H* requires a seal as shown at I to prevent pressure from escaping and also a seal as shown at *J* to keep dust and contamination out. Several designs are available. Some have a spring return, and no air leak whatsoever is possible.

The convolutions, which are in the radial space *K* between cylinder *C* and piston *A* is relatively very small, depending on the diameter of piston and thickness and type of material used for the diaphragm. Standard convolutions range from 1/16″ (or 1.5 mm) to 1/4″ (6 mm). The sidewall thickness of material ranges from 0.015″ (0.4 mm) to 0.035″ (0.9 mm). The diaphragms are usually fastened securely in place by providing a bead along the edge as shown at *L* and *M*. In other cases the air pressure will hold the diaphragm in position as shown at *N*.

Because the space *K* is very small, the tension on the side wall of the material is also very small. This tension is easily calculated by assuming the following:

S = stress on material, in pounds per inch of circumference
p = applied pressure, psi
K = width of space in inches

Since the stress S is calculated for 1″ of side wall material, the pressure sould be calculated for an area of 1 × K, so that:

$$S = \frac{p \times K \times 1}{2}$$

In the metric system the formula would be the same, with the following values:

S = stress on material in kg per centimeter of circumference
p = applied pressure, kg/cm^2
K = width of space in centimeters

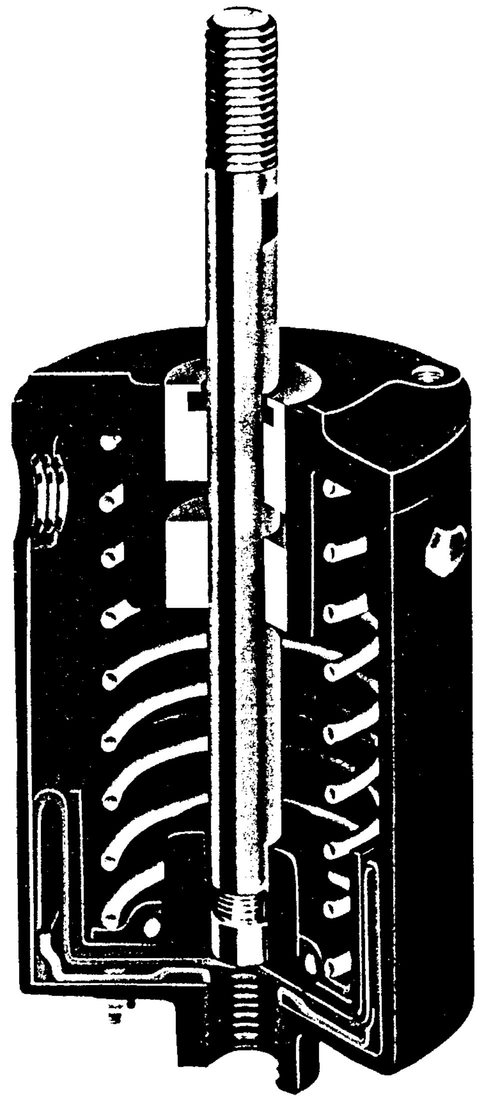
An Introduction to Pneumatic Systems

Anderson

Only very small amounts of leakage can be tolerated in the handling of dangerous fluids such as fluorine. In such cases either bellows or diaphragms are used. A diaphragm has a single involute and is limited in stroke. It also has special problems when the direction of pressure drop changes. Most diaphragms are molded to a shape near their normal operating shape. Consequently, a pressure reversal which causes the convolute to flip also causes excessive stress and shorter life and, of course, discontinuous actuation. The soft materials used in most diaphragms are not able to withstand either very high or very low temperatures. Metal bellows are often used where diaphragm stroke is inadequate or where the temperature causes diaphragm materials to fail.

Bellofram cylinders don't flip over at end of stroke.

"Prime" Mover Catalog



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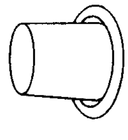


Diaphragms
As the leader in the industry we offer diaphragms in 15 standard elastomers and many special compounds which can withstand temperature ranges of −65°F to +600°F and pressures up to 3000 P.S.I.

Bellofram
CORPORATION
a Rexnord Company

The Bellofram Diaphragm Air Cylinder

The development of the long stroke rolling diaphragm for dynamic sealing proved to be the solution for many applications requiring low friction, no lubrication, low leakage, wide temperature variations, and low total cost. The popularity of the rolling diaphragm as a sealing means led to many requests for a standard line of "Off the Shelf" diaphragm cylinders; single and double acting, short and long stroke with a wide selection of effective areas. To meet these requests, the long stroke rolling diaphragm cylinder was developed and Bellofram has supplied many thousands of them since their 1965 introduction.

What are Diaphragm Air Cylinders?

Diaphragm Air Cylinders are actuators made from elastomers, engineered metals and fabrics. They require no lubrication, are virtually frictionless, and economical. They can be used to provide lifting, clamping, pushing, coining, turning, and other linear force or actuation motions in many applications.

Where are they used?

Diaphragm Air Cylinders are replacing conventionally sealed cylinders and actuators where low cost and reliability are requirements. They can be used with vacuum and gaseous pressure systems and applications are almost unlimited. They are currently solving many unique problems, being used as accumulators, pumps, reservoirs, expulsion chambers, shock mounts, impact absorbers, weld drivers, and tensioners.

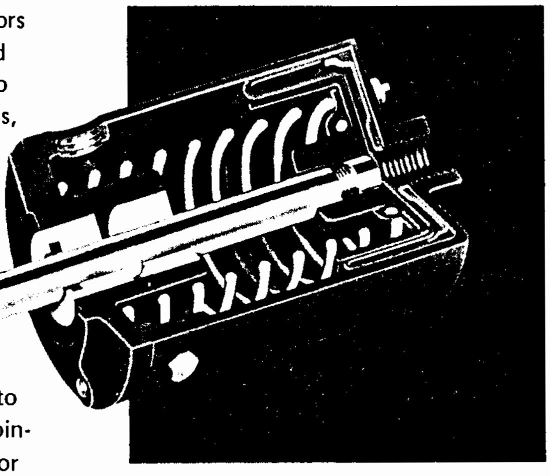
Why use Diaphragm Cylinders?

Use diaphragm cylinders when any or all of the following requirements must be met:

- No Lubrication
- Low Friction
- Extremely Sensitive Response To Small Pressure Variations
- Low Total Cost
- No Blow By Leakage
- Low Start-up Pressure

What materials are used in Diaphragm Air Cylinders?

The diaphragm is constructed of Neoprene® elastomer reinforced with Flex-Weave Dacron® fabric to give exceptionally long cycle life. The cylinder on sizes 4 and 6 is an impact-extruded aluminum shell. Larger sizes are made from a steel shell. The rods are ground, polished and hard-chrome plated steel. The bearings are sintered bronze, molybdenum disulphite impregnated. Other components are high strength materials with suitable corrosion resistant treatment.



Modifications

Our engineering department is continually providing products designed to meet specific customer needs. As a result, a variety of optional components are available, assuring you of the most versatile cylinder. The following are some of the more common options available upon special order:

Springs — A wide variety of special springs are available for any Bellofram air cylinder.

Rods — Rod materials, lengths, and end configurations can be adapted for special applications.

Shells — Special plating or painting of the cylinder shell is available when required.

Diaphragms — The Bellofram diaphragm can be made in an almost unlimited combination of elastomer and fabric. This would include such materials as Nitrile, Silicone, Fluorosilicone, elastomers, and various Dacron® and Nomex® fabrics.

Bearings — In addition to the linear ball bushing, a variety of bearings are available. This would include materials such as Teflon®/glass/Molybdenum (TGM), Celcon® and others.

The major areas for the use of modified cylinders are applications where the cylinder is in contact with corrosive materials, high temperature applications (up to 500°F) and in tension control applications where extreme sensitivity is required.

To date, well over 900 special modified cylinder designs have been produced. It is most likely that we have already modified a cylinder which will meet your needs.

General Operating Information

Bellofram Neoprene Diaphragm Air Cylinders are rated to operate on plant air up to 145 psi (10 bar) over temperatures from −40°F to +225°F.

Special diaphragm materials are available which permit our cylinders to operate at temperatures of −75°F to 400°F.

An air line lubricator is not necessary when operating a Bellofram air cylinder.

It is expected that the installation and operation procedures furnished with each cylinder will be followed for maximum service life.

External stroke limiters should be provided by the customer for limiting the stroke in both directions on single acting as well as double acting cylinders.

Hydraulic or Liquid Service

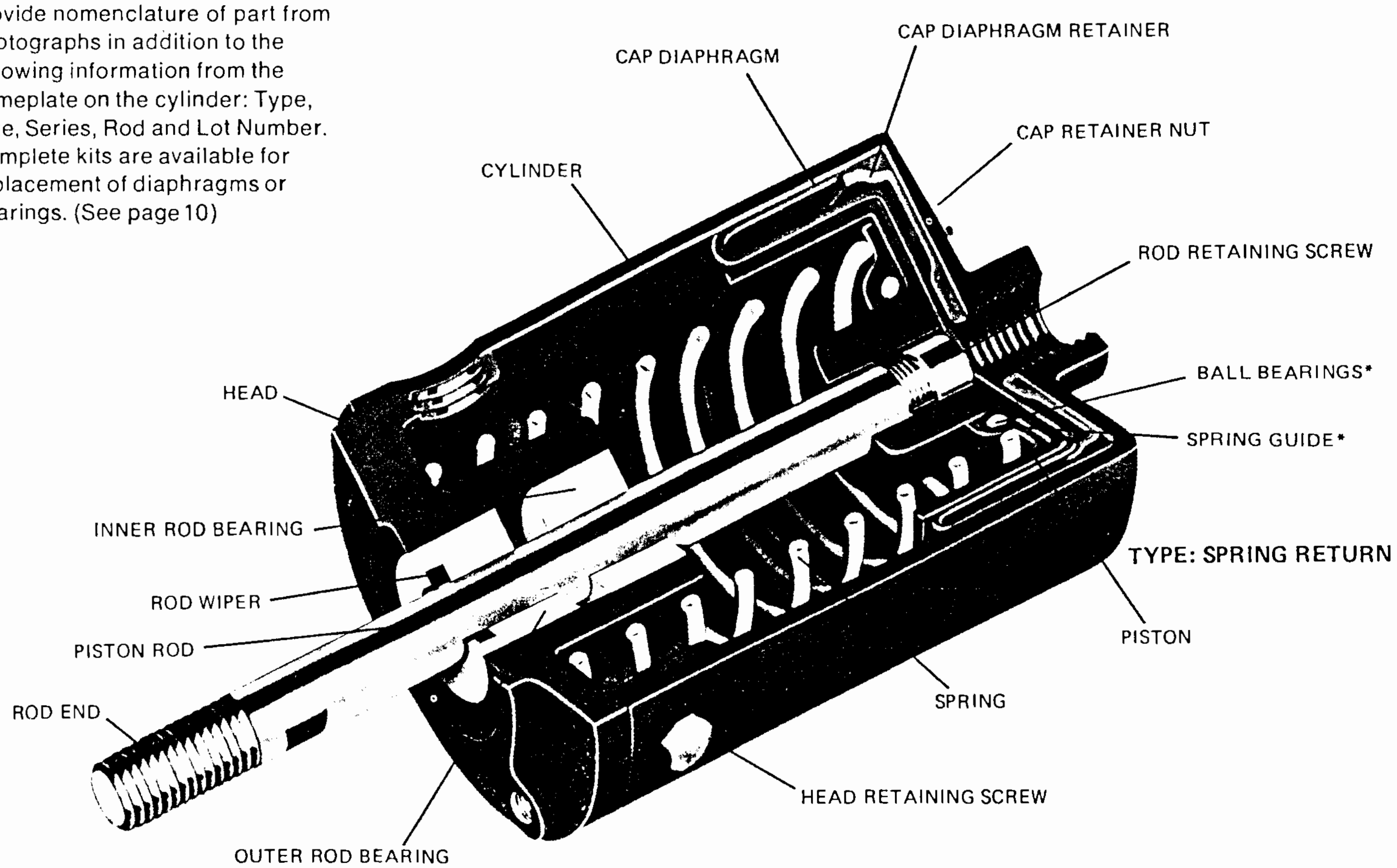
Actuating fluids other than air may be used by simply changing the diaphragm materials. Consult Bellofram's application engineering staff for information on hydraulic or liquid pressurized service.

Testing

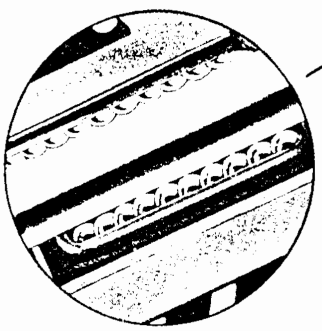
All cylinders are leak-tested prior to shipment. However, the cylinder is not a bubbletight assembly.

Nomenclature

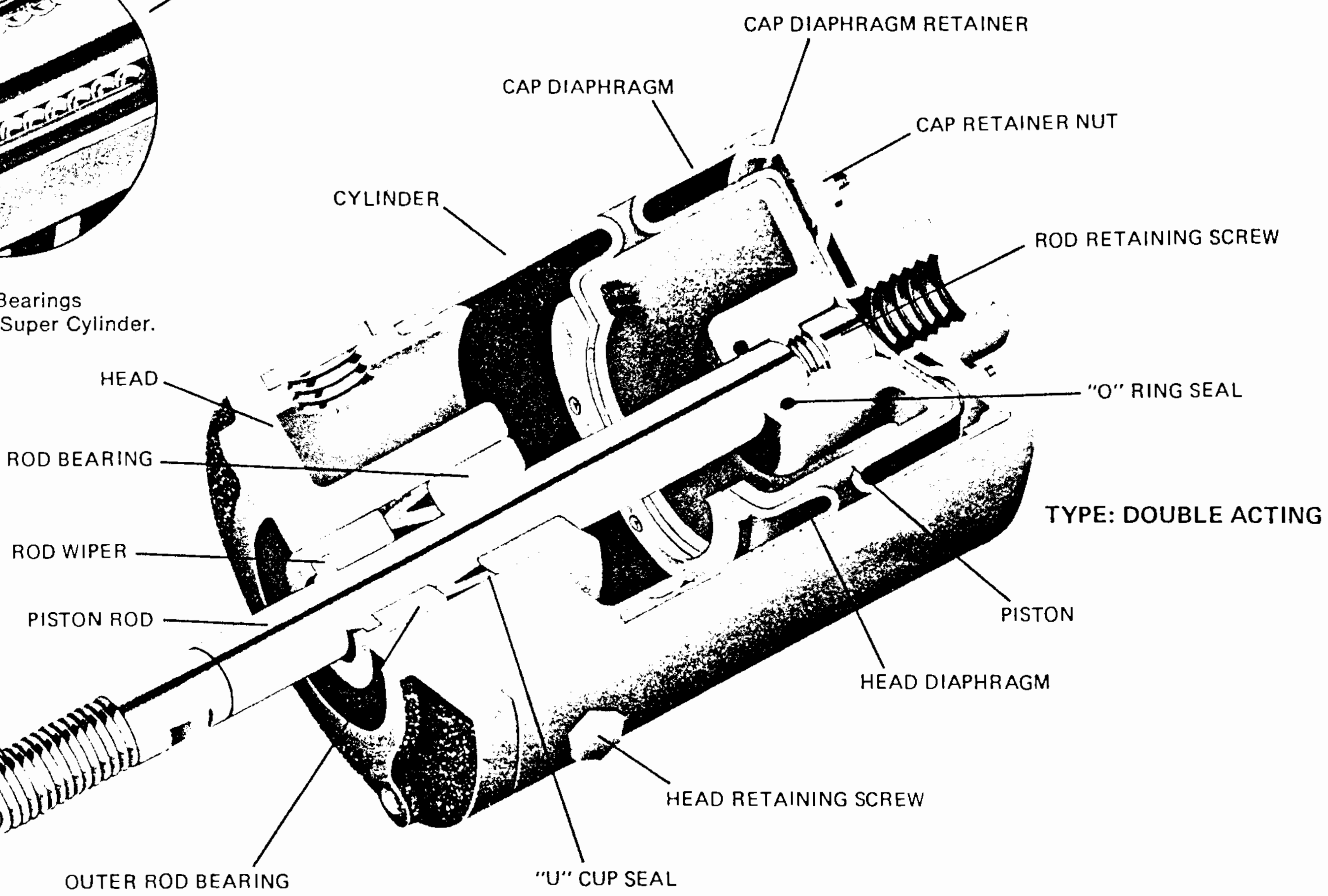
Ordering data for replacement parts
When ordering replacement parts provide nomenclature of part from photographs in addition to the following information from the nameplate on the cylinder: Type, Size, Series, Rod and Lot Number. Complete kits are available for replacement of diaphragms or bearings. (See page 10)



*Supplied on sizes 16F and larger.



Linear Ball Bearings available in Super Cylinder.



Cylinder Weights

Cyl. No.	Lbs.	Cyl. No.	Lbs.	Cyl. No.	Lbs.	Cyl. No.	Lbs.
S-4-F-BP	4	S-16-E-BP	13	D-4-F-BP	4	D-16-E-BP	14
S-4-BP-N	5	S-16-F-BP	14	D-4-N-BP	5	D-16-F-BP	16
S-6-F-BP	5	S-24-E-BP	18	D-6-F-BP	5	D-24-E-BP	20
S-6-BP-N	6	S-24-F-BP	25	D-6-N-BP	6	D-24-F-BP	28
S-9-F-BP	8	S-30-E-BP	25	D-9-F-BP	8	D-30-E-BP	26
S-9-BP-N	9	S-30-F-BP	31	D-9-N-BP	9	D-30-F-BP	33
S-12-E-BP	9	S-36-E-BP	28	D-12-E-BP	10	D-36-E-BP	29
S-12-F-BP	11	S-36-F-BP	36	D-12-F-BP	12	D-36-F-BP	39

Operating Data

Size (Effective Area) Sq. In.	Equiv. Bore Diam. In.	Spring Return						Double Acting	
		Stroke + .03 − .12 Series E In.	Stroke + .03 − .12 Series F In.	Approx. Spring Force — Zero Stroke (lbs.)		Approx. Increase Force Per In. of Stroke (lbs.)		Stroke + .03 − .12 Series E In.	Stroke + .03 − .12 Series F In.
				Series E	Series F	Series E	Series F		
4	2.3		1.80		6		3		1.3
6	2.8		2.40		9		4		1.9
9	3.4	2.20	3.00	17	12	4	4		2.5
12	3.9	2.30	3.60	18	18	6	6	1.8	3.1
16	4.5	2.62	4.20	24	24	8	8	2.1	3.7
24	5.5	2.60	5.24	36	36	11	11	2.0	4.6
30	6.3	3.07	6.00	45	54	13	14	2.4	5.4
36	6.8	3.55	6.00	54	54	16	14	2.9	5.4

Ordering Data

For: Universal, Foot, Clevis Bracket, Clevis Bracket Stud, Head Flange, Cap Flange and Nose Mounts.

D-12-F-BP-HFM

- Type: S — Spring Return, D — Double Acting
- Size: Area in sq. in. 4, 6, 9, 12, 16, 24, 30 or 36
- Mounting:
  - N — Nose Mount
  - UM — Universal Mount
  - FM — Foot Mount
  - CFM — Cap Flange Mount
  - HFM — Head Flange Mount
  - CBM — Clevis Bracket Mount
  - CBS — Clevis Bracket Stud
- Rod: BP — Bellofram Products Standard.
  SM — National Fluid Power Assoc. Standard
  Make selection from Dimensional Data Table
- Series: E or F Letter Determines Stroke
  Make selection from Operating Data Table

Example: D-12-F-BP-HFM is: a double acting, 12 sq. in., 3.1" stroke, BP rod end cylinder with a Head Flange Mount

For: Super Cylinder

All cylinders are "F" stroke single acting only.

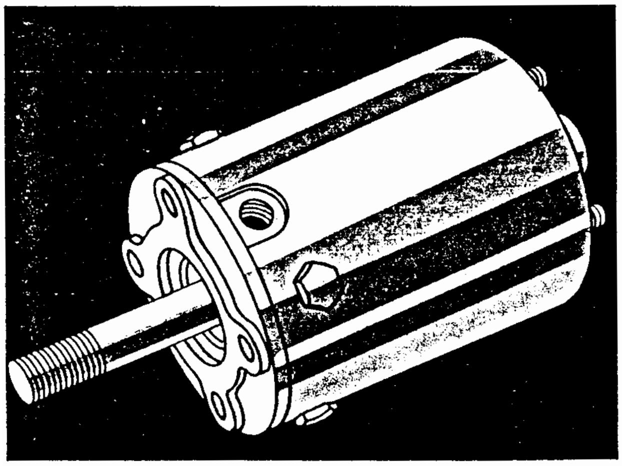
SS-12-F-SM-HFM

- Type: SS — Super Cylinder
- Size: Area in sq. in. 4, 6, 9, 12, 16, 24, 30 or 36
- Mounting:
  - UM — Universal Mount
  - FM — Foot Mount
  - CFM — Cap Flange Mount
  - HFM — Head Flange Mount
  - CBM — Clevis Bracket Mount
  - CBS — Clevis Bracket Stud
- Rod: BP — Bellofram Products Standard.
  SM — National Fluid Power Assoc. Standard
- Series: F Letter Determines Stroke
  Make selection from Operating Data Table

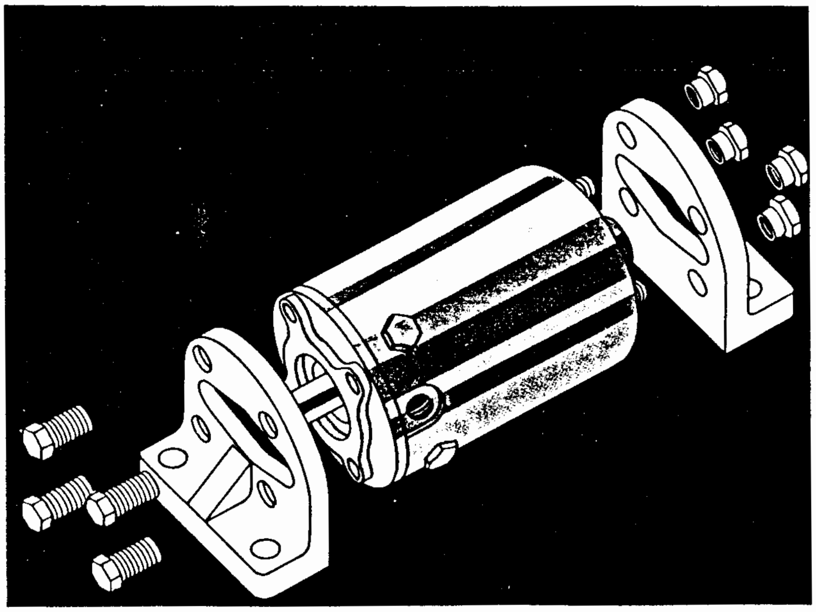
Example: SS-12-F-SM-HFM is: a single acting super cylinder, 12 sq. in., 3.6" stroke, SM rod end cylinder with a Head Flange Mount

STANDARD OPTIONS: NS — No Spring; NB — No Bearing. (Add to end of ordering data.)

Note: CAP MOUNTING STUDS WILL BE FURNISHED ONLY WHEN REQUESTED OR REQUIRED FOR MOUNTING ACCESSORIES ORDERED. Cap mounting stud data and dimensions are described on page 9.



UNIVERSAL MOUNT

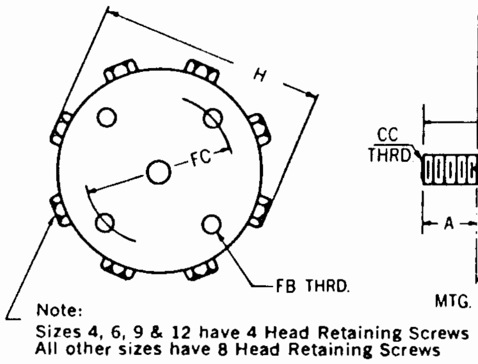


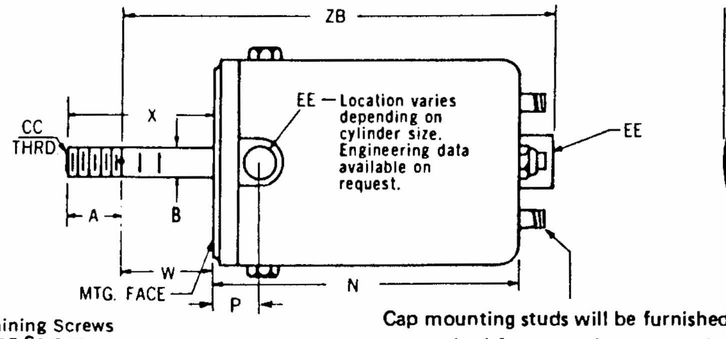
FOOT MOUNT

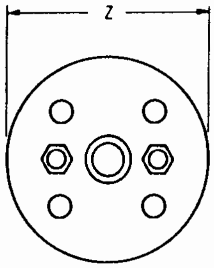
Universal Mount

Type: Spring Return or Double Acting. Stroke: Series E or F

145 psi service
(10 bar)







Cap mounting studs will be furnished only when requested or required for mounting accessories ordered. See ordering data on page 9.

Dimensional Data — Universal Mount (All dimensions in inches)

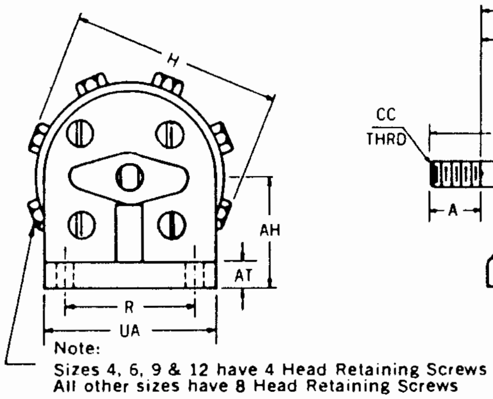
Size	Series	Z	H	N	EE	FC	FB	B	BP Rod End* X	BP A	BP W	BP ZB	BP CC	SM Rod End† X	SM A	SM W	SM ZB	SM CC	P
4	F	2.71	3.02	4.34	¼ NPT	2.00	¼-20	½	2.73	.75	1.98	6.72	⅜-24	2.73	1.00	1.73	6.47	5/16-20	.50
6	F	3.27	3.58	5.28	¼ NPT	2.00	¼-20	½	2.69	.75	1.94	7.63	⅜-24	2.69	1.00	1.69	7.38	5/16-20	.51
9	E	3.84	4.25	5.31	¼ NPT	3.00	7/16-14	¾	2.92	1.00	1.92	7.63	½-20	2.92	1.12	1.80	7.56	¾-16	.75
	F			6.34					2.69		1.69	8.44		2.69		1.57	8.32		
12	E	4.38	4.79	5.31	⅜ NPT	3.00	7/16-14	¾	2.92	1.00	1.92	7.78	½-20	2.92	1.12	1.80	7.66	¾-16	.75
	F			7.28					2.95		1.95	9.78		2.95		1.83	9.66		
16	E	4.99	5.40	6.03	⅜ NPT	3.00	½-13	¾	3.06	1.00	2.06	8.64	½-20	3.06	1.12	1.94	8.52	¾-16	.87
	F			8.38					2.78		1.78	10.71		2.78		1.56	10.59		
24	E	6.16	6.57	6.28	⅜ NPT	4.75	⅝-11	¾	2.86	1.00	1.86	8.73	½-20	2.86	1.12	1.74	8.59	¾-16	1.00
	F			10.22					2.44		1.44	12.08		2.44		1.32	12.03		
30	E	6.88	7.29	7.00	⅜ NPT	4.75	⅝-11	1	2.83	1.25	1.58	9.26	⅝-18	2.83	1.50	1.33	9.30	1-14	1.00
	F			11.44					3.05	1.50	1.55	13.53	1-12	3.05		1.55	13.53		
36	E	7.38	7.79	7.69	⅜ NPT	4.75	⅝-11	1	2.83	1.25	1.58	9.82	⅝-18	2.83	1.50	1.33	10.00	1-14	1.00
	F			11.47					3.05	1.50	1.55	13.54	1-12	3.05		1.55	13.54		

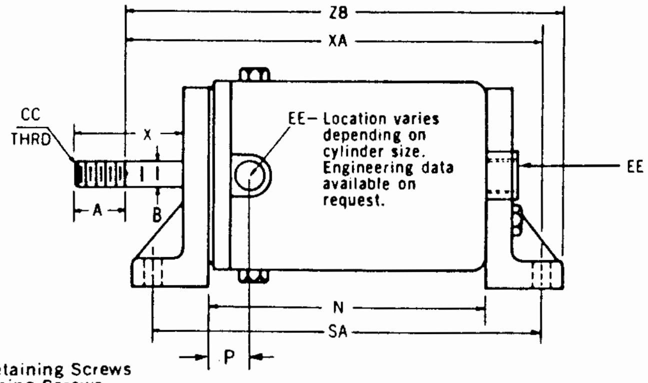
*BP Rod End — Bellofram Products Co. Standard
†SM Rod End — National Fluid Power Assoc. Standards

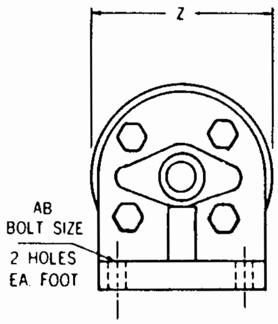
Foot Mount

Type: Spring Return or Double Acting. Stroke: Series E or F

145 psi service
(10 bar)







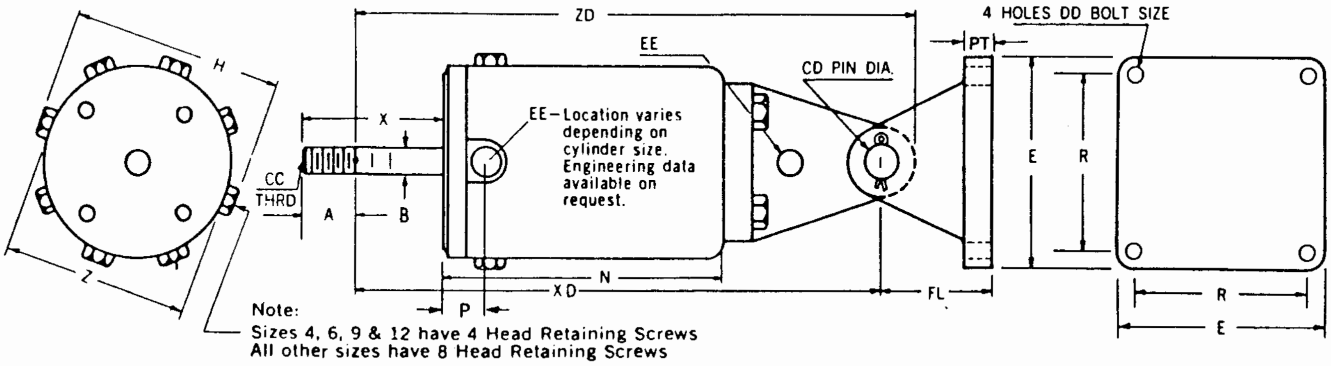
Dimensional Data — Foot Mount (All dimensions in inches)

Size	Series	Z	H	N	EE	B	BP Rod End* X	BP A	BP XA	BP ZB	BP CC	SM Rod End† X	SM A	SM XA	SM ZB	SM CC	AT	AH	UA	R	SA	AB	P
4	F	2.71	3.02	4.34	¼ NPT	½	2.41	.75	7.44	7.82	⅜-24	2.41	1.00	7.19	7.57	5/16-20	.38	1.88	2.62	2.00	6.59	¼	.50
6	F	3.27	3.58	5.28	¼ NPT	½	2.38	.75	8.35	8.72	⅜-24	2.38	1.00	8.10	8.47	5/16-20	.38	1.88	2.62	2.00	7.53	¼	.51
9	E	3.84	4.15	5.31	¼ NPT	¾	2.24	1.00	8.86	9.55	½-20	2.24	1.12	8.74	9.43	¾-16	.56	2.75	4.00	3.00	8.56	½	.75
	F			6.34			2.06		9.67	10.29		2.06		9.55	10.16						9.59		
12	E	4.38	4.79	5.31	⅜ NPT	¾	2.30	1.00	8.86	9.56	½-20	2.30	1.12	8.73	9.44	¾-16	.56	2.75	4.00	3.00	8.56	½	.75
	F			7.28			2.33		10.86	11.56		2.33		10.73	11.36						10.53		
18	E	4.99	5.40	6.03	⅜ NPT	¾	2.31	1.00	9.72	10.42	½-20	2.34	1.12	9.59	10.22	¾-16	.58	2.75	4.00	3.00	9.28	½	.87
	F			8.38			2.16		11.78	12.41		2.16		11.66	12.28						11.62		
24	E	6.16	6.57	6.28	⅜ NPT	¾	2.23	1.00	10.09	10.86	½-20	2.23	1.12	9.94	10.63	¾-16	.68	4.00	6.25	4.75	10.16	⅝	1.00
	F			10.22			1.80		13.06	14.29		1.80		13.10	13.79						14.09		
30	E	6.88	7.29	7.00	⅜ NPT	1	2.20	1.25	10.52	11.20	⅝-18	2.20	1.50	10.26	10.95	1-14	.68	4.00	6.25	4.75	10.88	⅝	1.00
	F			11.44			2.41	1.50	14.92	15.61	1-12	2.41		14.92	15.69						15.31		
36	E	7.38	7.79	7.69	⅜ NPT	1	2.20	1.25	11.22	11.91	⅝-18	2.20	1.50	10.97	11.66	1-14	.68	4.00	6.25	4.75	11.56	⅝	1.00
	F			11.47			2.41	1.50	14.94	15.62	1-12	2.41		14.94	15.62						15.34		

*BP Rod End — Bellofram Products Co. Standard
†SM Rod End — National Fluid Power Assoc. Standards

Clevis Bracket Mount (or Stud)** Type: Spring Return or Double Acting. Stroke: Series E or F

145 psi service (10 bar)



Note:
Sizes 4, 6, 9 & 12 have 4 Head Retaining Screws
All other sizes have 8 Head Retaining Screws

Dimensional Data — Clevis Bracket Mount (All dimensions in inches)

Size	Series	Z	H	N	EE	B	BP Rod End* X	BP Rod End* A	BP Rod End* XD	BP Rod End* ZD	BP Rod End* CC	SM Rod End† X	SM Rod End† A	SM Rod End† XD	SM Rod End† ZD	SM Rod End† CC	CD	DD	R	E	FL	EW	PT	P
4	F	2.71	3.02	4.34	¼ NPT	½	2.73	.75	8.45	9.07	⅜-24	2.73	1.00	8.19	8.82	7/16-20	.625	¼	2.38	3.12	1.38	.93	⅜	.50
6	F	3.27	3.58	5.28	¼ NPT	½	2.69	.75	9.35	9.97	⅜-24	2.69	1.00	9.09	9.72	7/16-20	.625	¼	2.38	3.12	1.38	.93	½	.51
9	E	3.84	4.25	5.31	¼ NPT	¾	2.92	1.00	9.98	10.73	½-20	2.92	1.12	9.86	10.61	¾-16	.750	¼	3.00	3.75	1.69	.99	½	.75
	F			6.34			2.69		10.80	11.55		2.69		10.67	11.42									
12	E	4.38	4.79	5.31	⅜ NPT	¾	2.92	1.00	10.23	10.98	½-20	2.92	1.12	10.11	10.86	¾-16	.750	⅜	3.00	4.00	1.75	1.24	½	.75
	F			7.28			2.95		12.23	12.98		2.95		12.11	12.86									
16	E	4.99	5.40	6.03	⅜ NPT	¾	3.06	1.00	11.09	11.84	½-20	3.06	1.12	10.97	11.72	¾-16	.750	½	3.00	4.00	1.75	1.24	½	.87
	F			8.38			2.78		13.16	13.91		2.78		13.03	13.78									
24	E	6.16	6.57	6.28	⅜ NPT	¾	2.86	1.00	10.78	11.78	½-20	2.86	1.12	11.16	12.16	¾-16	1.000	½	4.00	5.12	2.00	1.49	⅝	1.00
	F			10.22			2.44		15.22	16.22		2.44		14.68	15.68									
30	E	6.88	7.29	7.00	⅜ NPT	1	2.83	1.25	11.70	12.70	⅝-18	2.83	1.50	11.89	12.89	1-14	1.000	½	4.00	5.12	2.00	1.49	⅝	1.00
	F			11.44			3.05	1.50	16.11	17.11	1-12	3.05		16.11	17.11									
36	E	7.38	7.79	7.69	⅜ NPT	1	2.83	1.25	12.31	13.31	⅝-18	2.83	1.50	12.60	13.60	1-14	1.000	½	4.00	5.12	2.00	1.49	⅝	1.00
	F			11.47			3.05	1.50	16.13	17.13	1-12	3.05		16.13	17.13									

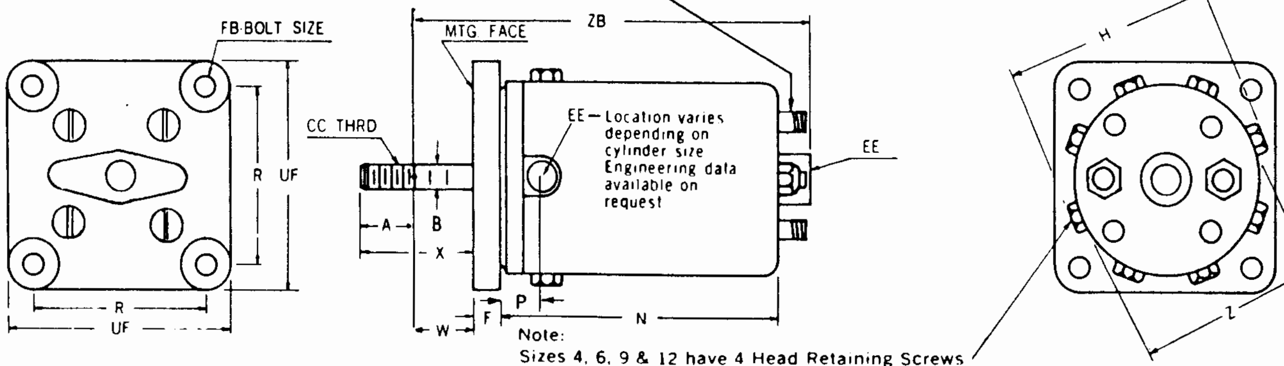
* BP Rod End — Bellofram Products Co. Standard
** See note under Clevis Bracket Mount illustration on next page
† SM Rod End — National Fluid Power Assoc. Standards

Head Flange Mount Type: Spring Return or Double Acting. Stroke: Series E or F

145 psi service (10 bar)

Cap mounting stud data and dimensions available on request.

Cap mounting studs will be furnished only when requested or required for mounting accessories ordered.



Note:
Sizes 4, 6, 9 & 12 have 4 Head Retaining Screws
All other sizes have 8 Head Retaining Screws

Dimensional Data — Head Flange Mount (All dimensions in inches)

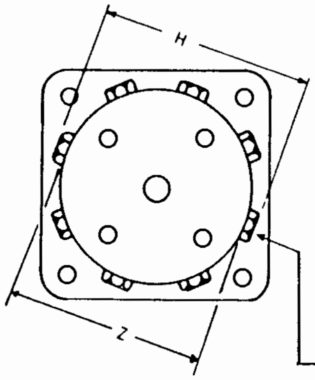
Size	Series	Z	H	N	EE	B	BP Rod End* X	BP Rod End* A	BP Rod End* W	BP Rod End* ZB	BP Rod End* CC	SM Rod End† X	SM Rod End† A	SM Rod End† W	SM Rod End† ZB	SM Rod End† CC	F	R	UF	FB	P
4	F	2.71	3.02	4.34	¼ NPT	½	1.95	.75	1.20	6.72	⅜-24	1.95	1.00	0.95	6.47	7/16-20	.781	2.81	3.62	¼	.50
6	F	3.27	3.58	5.28	¼ NPT	½	1.91	.75	1.16	7.63	⅜-24	1.91	1.00	0.91	7.38	7/16-20	.781	2.81	3.62	¼	.51
9	E	3.84	4.15	5.31	¼ NPT	¾	2.23	1.00	1.23	7.63	½-20	2.23	1.12	1.11	7.56	¾-16	.690	4.38	5.50	5/16	.75
	F			6.34			2.00		1.00	8.44		2.00		0.88	8.32						
12	E	4.38	4.79	5.31	⅜ NPT	¾	2.23	1.00	1.23	7.78	½-20	2.23	1.12	1.11	7.66	¾-16	.690	4.38	5.50	5/16	.75
	F			7.28			2.26		1.26	9.78		2.26		1.14	9.66						
16	E	4.99	5.40	6.03	⅜ NPT	¾	2.37	1.00	1.36	8.64	½-20	2.37	1.12	1.25	8.52	¾-16	.690	4.38	5.50	½	.87
	F			8.38			2.09		1.09	10.71		2.09		0.97	10.59						
24	E	6.16	6.57	6.28	⅜ NPT	¾	2.17	1.00	1.17	8.73	½-20	2.17	1.12	1.05	8.59	¾-16	.656	6.00	7.50	½	1,00
	F			10.22			1.78		0.78	12.08		1.75		0.63	12.03						
30	E	6.88	7.29	7.00	⅜ NPT	1	2.14	1.25	0.99	9.26	⅝-18	2.14	1.50	0.64	9.30	1-14	.656	6.00	7.50	½	1.00
	F			11.44			2.36	1.50	0.86	13.53	1-12	2.36		0.86	13.53						
36	E	7.38	7.79	7.69	⅜ NPT	1	2.14	1.25	0.99	9.82	⅝-18	2.14	1.50	0.64	10.00	1-14	.656	6.00	7.50	½	1.00
	F			11.47			2.36	1.50	0.86	13.54	1-12	2.36		0.86	13.69						

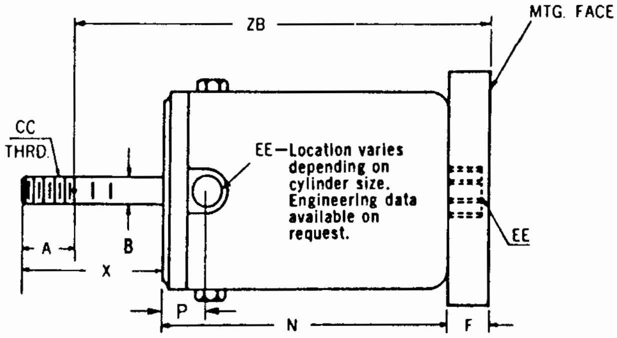
*BP Rod End — Bellofram Products Co. Standard
†SM Rod End — National Fluid Power Assoc. Standards

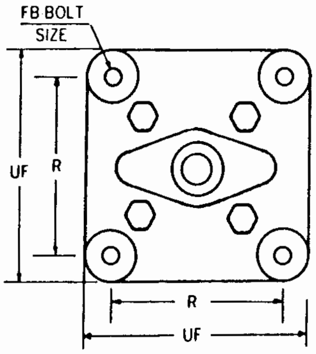
Cap Flange Mount

Type: Spring Return or Double Acting. Stroke: Series E or F

145 psi service (10 bar)







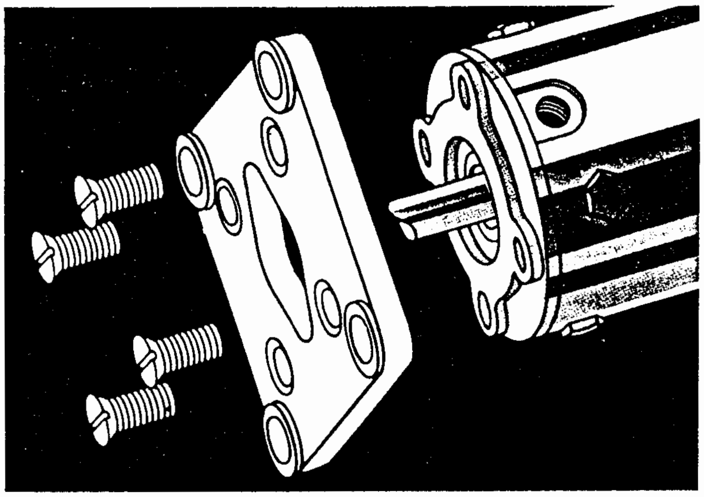
Note:
Sizes 4, 6, 9 & 12 have 4 Head Retaining Screws
All other sizes have 8 Head Retaining Screws

Dimensional Data — Cap Flange Mount (All dimensions in inches)

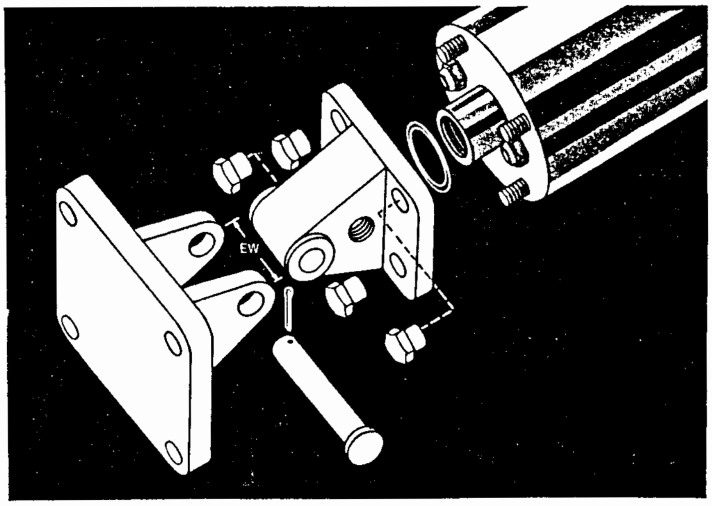
Size	Series	Z	H	N	EE	B	BP Rod End*				SM Rod End†				F	R	UF	FB	P
							X	A	ZB	CC	X	A	ZB	CC					
4	F	2.71	3.02	4.34	¼ NPT	½	2.73	.75	7.10	⅜-24	2.73	1.00	6.85	7/16-20	.781	2.81	3.62	¼	.50
6	F	3.27	3.58	5.28	¼ NPT	½	2.69	.75	8.00	⅜-24	2.69	1.00	7.75	7/16-20	.781	2.81	3.62	¼	.51
9	E	3.84	4.15	5.31	¼ NPT	¾	2.92	1.00	7.92	½-20	2.92	1.12	7.80	¾-16	.690	4.38	5.50	7/16	.75
	F			6.34			2.69		8.72		2.69		8.60						
12	E	4.38	4.79	5.31	⅜ NPT	¾	2.92	1.00	7.92	½-20	2.92	1.12	7.80	¾-16	.690	4.38	5.50	7/16	.75
	F			7.28			2.95		9.92		2.95		9.80						
16	E	4.99	5.40	6.03	⅜ NPT	¾	3.06	1.00	8.18	½-20	3.06	1.12	8.66	¾-16	.690	4.38	5.50	½	.87
	F			8.38			2.78		10.85		2.78		10.73						
24	E	6.16	6.57	6.28	⅜ NPT	¾	2.86	1.00	8.53	½-20	2.86	1.12	8.71	¾-16	.690	6.00	7.50	½	1.00
	F			10.22			2.44		12.35		2.44		12.23						
30	E	6.88	7.29	7.00	⅜ NPT	1	2.83	1.25	9.27	⅝-18	2.83	1.50	8.02	1-14	.690	6.00	7.50	½	1.00
	F			11.44			3.05	1.50	13.68	1.12	3.05		13.68						
36	E	7.38	7.79	7.69	⅜ NPT	1	2.83	1.25	9.96	⅝-18	2.83	1.50	9.71	1-14	.690	6.00	7.50	½	1.00
	F			11.47			3.05	1.50	13.71	1.12	3.05	1.50	13.71						

*BP Rod End — Bellofram Products Co. Standard
†SM Rod End — National Fluid Power Assoc. Standards

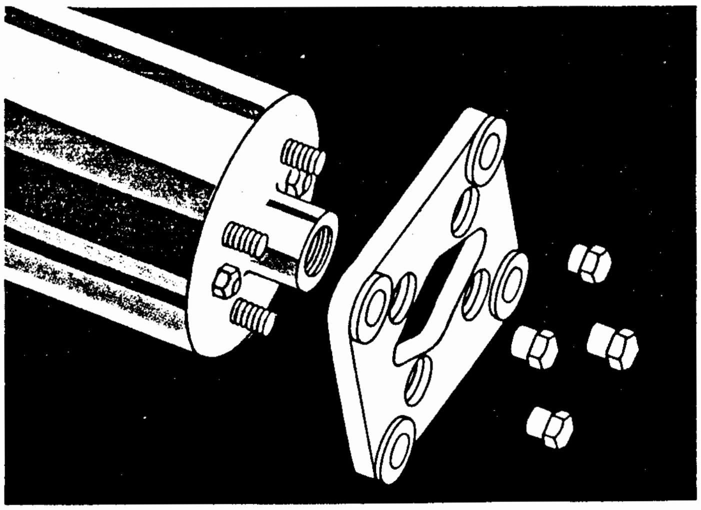
HEAD FLANGE MOUNT



CLEVIS BRACKET MOUNT



Clevis Bracket Mount includes parts as shown in white. Clevis Bracket Stud includes only the Male Bracket, O-Ring and four Sleeve Nuts. Note: Refer to top of opposite page for "EW" dimensions.

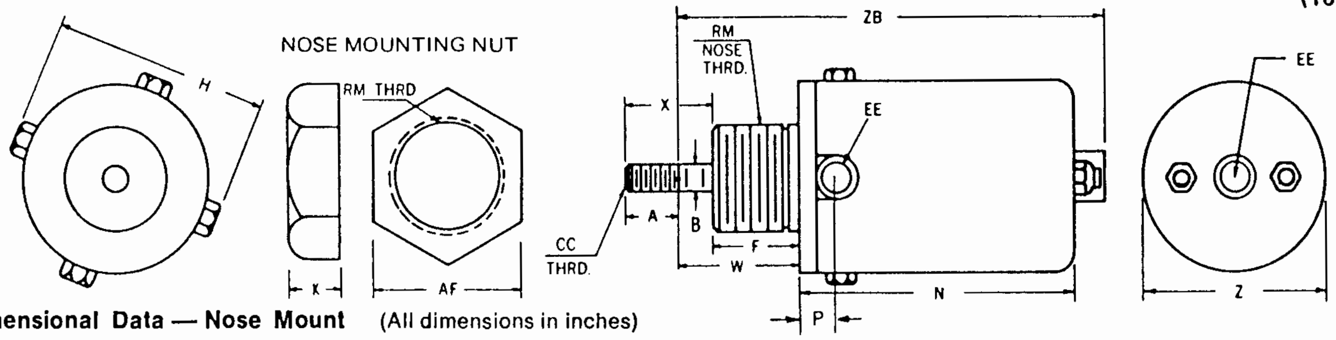


CAP FLANGE MOUNT

Nose Mount

Type: Spring Return or Double Acting Stroke: Series E or F

145 psi service
(10 bar)



Dimensional Data — Nose Mount (All dimensions in inches)

Size	Series	Z	H	N	EE	F	RM	B	BP Rod End* X	BP Rod End* A	BP Rod End* W	BP Rod End* ZB	BP Rod End* CC	SM Rod End† X	SM Rod End† A	SM Rod End† W	SM Rod End† ZB	SM Rod End† CC	AF	K	P
4	F	2.71	3.02	4.34	1/4 NPT	1.25	1 3/8-12	1/2	1.48	.75	1.98	6.72	3/8-24	1.48	1.00	1.73	6.47	7/16-20	2.06	.78	.59
6	F	3.27	3.58	5.28	1/4 NPT	1.25	1 3/8-12	1/2	1.44	.75	1.94	7.63	3/8-24	1.44	1.00	1.69	7.06	7/16-20	2.06	.78	.59
9	E / F	3.84	4.15	5.16 / 6.19	1/4 NPT	1.25	1 5/8-12	3/4	1.83 / 1.61	1.00	2.08 / 1.86	7.65 / 8.45	1/2-20	1.83 / 1.61	1.12	1.96 / 1.74	7.53 / 8.33	3/4-16	2.44	.91	.59

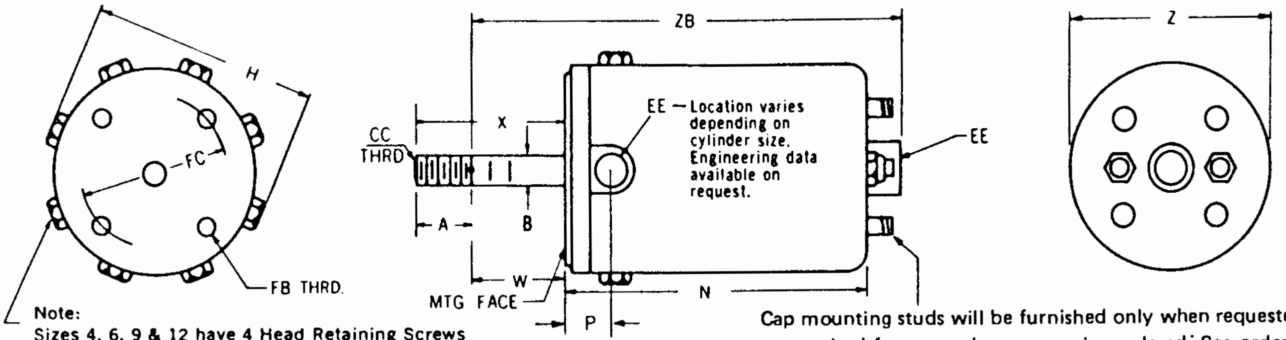
*BP Rod End — Bellofram Products Co. Standard †SM Rod End — National Fluid Power Assoc. Standards

Super Cylinder

145 psi service
(10 bar)

For applications where a maximum of sensitivity is required, we now offer the "Super Cylinder".

The super sensitivity of this model is made possible by the incorporation of a linear ball bearing and hardened steel rod. Friction is reduced to an absolute minimum, making the Super Cylinder virtually frictionless.

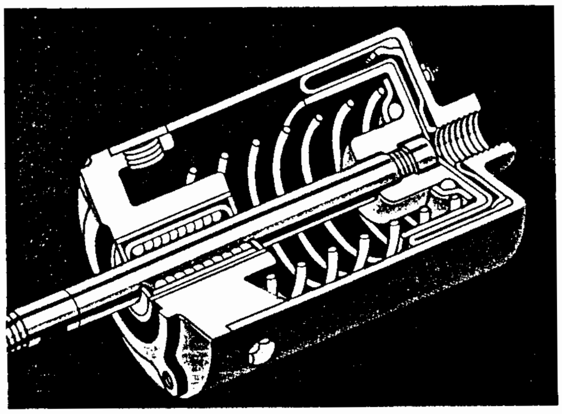


Note:
Sizes 4, 6, 9 & 12 have 4 Head Retaining Screws
All other sizes have 8 Head Retaining Screws

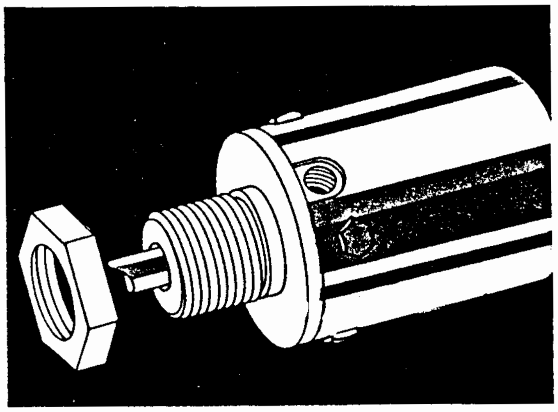
Cap mounting studs will be furnished only when requested or required for mounting accessories ordered. See ordering data, next page.

Dimensional Data — Super Cylinder (All dimensions in inches)

Size	Z	H	N	EE	FC	FB	B	BP Rod End X	BP Rod End A	BP Rod End W	BP Rod End ZB	BP Rod End CC	SM Rod End X	SM Rod End A	SM Rod End W	SM Rod End ZB	SM Rod End CC	P
4	2.71	3.02	4.34	1/4 NPT	2.00	1/4-20	1/2	3.10	.75	2.35	7.09	3/8-24	3.10	1.00	2.10	6.85	7/16-20	.50
6	3.27	3.58	5.28	1/4 NPT	2.00	1/4-20	1/2	2.16	.75	1.41	7.09	3/8-24	2.16	1.00	1.16	6.85	7/16-20	.51
9	3.84	4.25	6.34	1/4 NPT	3.00	7/16-14	3/4	3.50	1.00	2.50	9.25	1/2-20	3.50	1.12	2.38	9.13	3/4-16	.75
12	4.38	4.79	7.28	3/8 NPT	3.00	7/16-14	3/4	2.57	1.00	1.57	9.38	1/2-20	2.57	1.12	1.45	9.26	3/4-16	.75
16	4.99	5.40	8.38	3/8 NPT	3.00	1/2-13	3/4	3.78	1.00	2.78	11.69	1/2-20	3.78	1.12	2.66	11.57	3/4-16	.87
24	6.16	6.57	10.22	3/8 NPT	4.75	5/8-11	3/4	2.00	1.00	1.00	11.75	1/2-20	2.00	1.12	1.00	11.75	3/4-16	1.00
30	6.88	7.29	11.44	3/8 NPT	4.75	5/8-11	1	3.05	1.50	1.55	13.52	1-12	3.05	1.50	1.55	13.52	1-14	1.00
36	7.38	7.79	11.47	3/8 NPT	4.75	5/8-11	1	3.05	1.50	1.55	13.55	1-12	3.05	1.50	1.55	13.55	1-14	1.00



SUPER CYLINDER

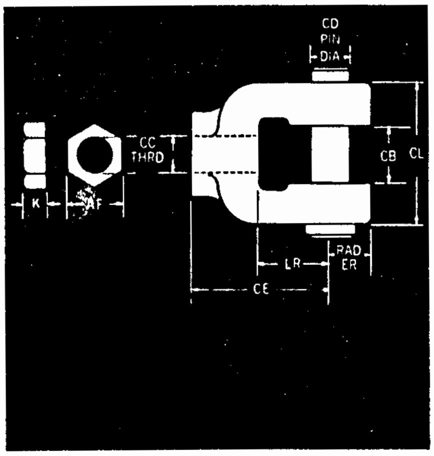


NOSE MOUNT

Rod Clevis and Jam Nut

Size	BP* Rod End CC Thrd.	SM† Rod End CC Thrd.	Series	CB	CD Pin Dia.	CE	CL	ER Rad.	LR	AF	K
4	3/8-24	–	F	.56	3/8	1 7/8	1.38	.53	1.25	9/16	7/32
	–	7/16-20	F	.56	7/16	1 7/8	1.38	.53	1.25	11/16	1/4
6	3/8-24	–	F	.56	3/8	1 7/8	1.38	.53	1.25	9/16	7/32
	–	7/16-20	F	.56	7/16	1 7/8	1.38	.53	1.25	11/16	1/4
9	1/2-20	–	E	.56	1/2	1 7/8	1.38	.53	1.25	3/4	5/16
	1/2-20	–	F	.56	1/2	1 7/8	1.38	.53	1.25	3/4	5/16
	–	3/4-16	E	.88	3/4	2 3/8	2.12	.75	1.25	1 1/8	27/64
	–	3/4-16	F	.83	3/4	2 3/8	2.12	.75	1.25	1 1/8	27/64
12	1/2-20	–	E	.56	1/2	1 7/8	1.38	.53	1.25	3/4	5/16
	1/2-20	–	F	.56	1/2	1 7/8	1.38	.53	1.25	3/4	5/16
	–	3/4-16	E	.88	3/4	2 3/8	2.12	.75	1.25	1 1/8	27/64
	–	3/4-16	F	.88	3/4	2 3/8	2.12	.75	1.25	1 1/8	27/64
16	1/2-20	–	E	.56	1/2	1 7/8	1.38	.53	1.25	3/4	5/16
	1/2-20	–	F	.56	1/2	1 7/8	1.38	.53	1.25	3/4	5/16
	–	3/4-16	E	.88	3/4	2 3/8	2.12	.75	1.25	1 1/8	27/64
	–	3/4-16	F	.88	3/4	2 3/8	2.12	.75	1.25	1 1/8	27/64
24	1/2-20	–	E	.56	1/2	1 7/8	1.38	.53	1.25	3/4	5/16
	1/2-20	–	F	.56	1/2	1 7/8	1.38	.53	1.25	3/4	5/16
	–	3/4-16	E	.88	3/4	2 3/8	2.12	.75	1.25	1 1/8	27/64
	–	3/4-16	F	.88	3/4	2 3/8	2.12	.75	1.25	1 1/8	27/64
30	5/8-18	–	E	.56	9/16	1 7/8	1.38	.53	1.25	15/16	3/8
	1-12	–	F	1.50	1	3 1/2	3.50	1.00	1.50	1 1/2	35/64
	–	1-14	E	1.50	1	3 1/2	3.50	1.00	1.50	1 1/2	35/64
	–	1-14	F	1.50	1	3 1/2	3.50	1.00	1.50	1 1/2	35/64
36	5/8-18	–	E	.56	9/16	1 7/8	1.38	.53	1.25	15/16	3/8
	1-12	–	F	1.50	1	3 1/2	3.50	1.00	1.50	1 1/2	35/64
	–	1-14	E	1.50	1	3 1/2	3.50	1.00	1.50	1 1/2	35/64
	–	1-14	F	1.50	1	3 1/2	3.50	1.00	1.50	1 1/2	35/64

*BP Rod End — Bellofram Products Co. Standard
†SM Rod End — National Fluid Power Assoc. Standards



Ordering Data

36-F-BP-RC

- Size: Area in sq. in. 4, 6, 9, 12, 16, 24, 30 or 36
- Series: E or F Letter Determines Stroke
  Make selection from Operating Data Table

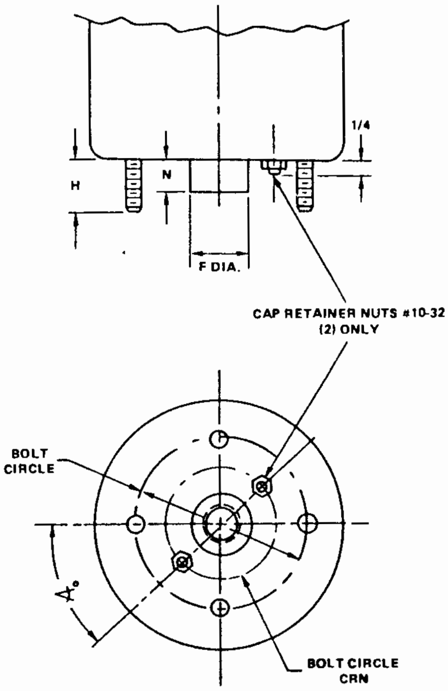
} From cylinder on which clevis is to be used.

- Rod Clevis
- Rod: BP — Bellofram Products Standard
  SM — National Fluid Power Assn. Standard
  Make selection from Dimensional Data Table

Example: 36-F-BP-RC is a Rod Clevis for a size 36 sq. in. cylinder with a 1-12 rod thread.

Cylinder Cap End Mounting Studs

Size	No. of Studs	Size of Stud	H (Approx.)	Bolt Circle	F (Boss)	N	Bolt Circle CRN	∢°
4	2	1/4-20	5/8	1-5/16	11/16	13/32	1-3/8	90
6	2	1/4-20	5/8	1-1/2	11/16	13/32	1-3/8	90
9	4	1/4-20	5/8	2	11/16	13/32	1-3/8	45
12	4	1/4-20	5/8	2-5/16	1	17/32	1-11/16	45
16	4	3/8-16	5/8	2-5/16	1	17/32	1-11/16	45
24	4	3/8-16	9/16	3-1/8	1	17/32	1-11/16	45
30	4	1/2-13	11/16	4	1	17/32	1-11/16	45
36	4	1/2-13	11/16	4	1	17/32	1-11/16	45



Repair Kits

The following is included in the repair kits:

Spring Return Diaphragm Kit

1. Diaphragm, Cap
2. Retainer, Adhesive, Cap
3. Nuts, Cap Retainer
4. Instructions

Spring Return Bearing Kit

1. Inner Bearing
2. Outer Bearing
3. Rod Wiper
4. Instructions

Double Acting Diaphragm Kit

1. Diaphragm, Cap
2. Diaphragm, Head
3. Retainer, Adhesive, Cap
4. Retainer, Adhesive, Head
5. Rivets, Blind (or screws)
6. Nuts, Cap Retainer
7. Seal "O" Ring
8. Instructions

Double Acting Bearing Kit

1. Inner Bearing
2. Outer Bearing
3. Rod Wiper
4. U-Cup Seal
5. Instructions

Repair Kits are available to permit user in-plant maintenance without delay and expense of returning parts to the factory. Each kit includes installation instructions. Nameplate data of the cylinder must accompany order to insure receipt of correct parts.

<table>
<tr><th colspan="4">Kits For Spring Return Cylinders</th><th colspan="4">Kits For Double Acting Cylinders</th></tr>
<tr><th>For Models</th><th>Diaphragm Kit No.</th><th>For Models</th><th>Bearing Kit No.</th><th>For Models</th><th>Diaphragm Kit No.</th><th>For Models</th><th>Bearing Kit No.</th></tr>
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S-36-F-SM</td><td>S36F</td><td></td><td></td></tr>
</table>

Ordering Data
Order by kit number above

Breather Vents

Breather vents are available for use on Bellofram Spring Return Air Cylinders.

The Breather, which contains a 40 micron bronze filter, is simply threaded into the air relief port of the cylinder head. It prevents foreign matter from being drawn into the cylinder on the return stroke of the piston, and also acts as a snubber. The snubbing reduces the piston speed and impact at the end of the stroke in both directions.

Ordering Data
Breather Vent for ¼" pipe tap
(Fits cylinder sizes -4, -6, -9)
Part No. 315-661-001
(Formerly #BV-2)

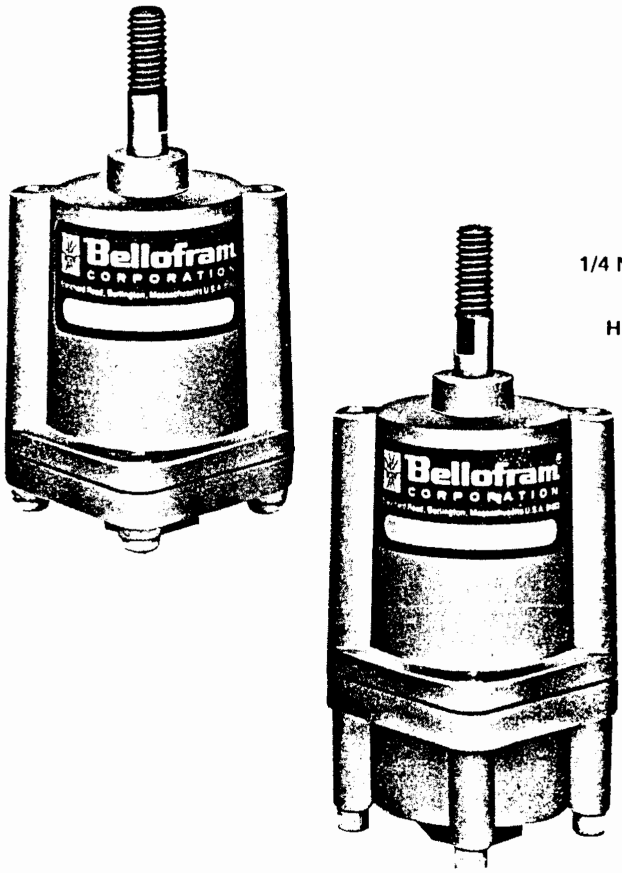
Breather Vent for ⅜" pipe tap
(Fits cylinder sizes -12, -16, -24, -30, -36)
Part No. 315-661-002
(Formerly #BV-3)

Warranty

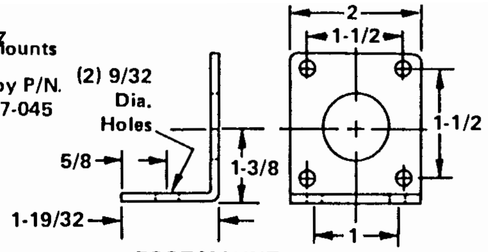
Bellofram Corporation guarantees the products of its manufacture to be free of defects of materials and workmanship in normal use for a period of ninety (90) days from date of sale to customer. The guarantee is limited to repair or replacement of the defective product at the exclusive option of Bellofram. In the event a claim should arise during the guarantee period, Bellofram must be notified immediately and the defective product made available for field or factory inspection and disposition. Bellofram cannot and does not accept responsibility of any type for any of its products that have been subjected to improper installation, application, negligence, tampering or abuse. In any event, the liability of Bellofram for a defective product is limited to the cost of the product or a replacement therefor. This warranty is in lieu of any and all other warranties, express or implied. Bellofram Corporation does not warrant its products to be merchantable and does not warrant its products in any manner for any specific purpose or use, and Bellofram Corporation disclaims any liability for consequential damages of any nature.

Prices and specifications are subject to change without notice.

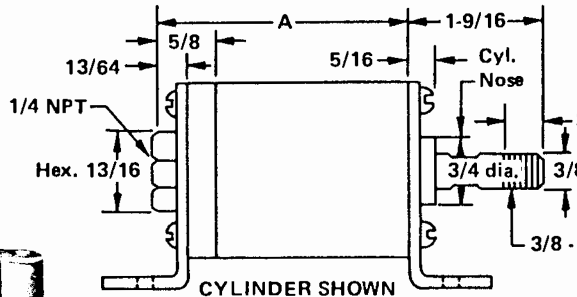
Low-Cost Cylinders

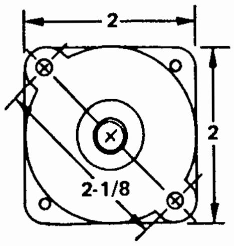


Foot Mounts
Order by P/N.
414-607-045



FOOT MOUNT BRACKET DIMENSIONS



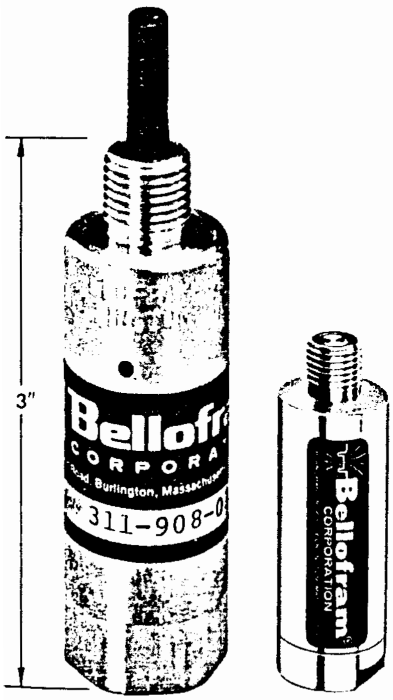


CYLINDER SHOWN WITH OPTIONAL FOOT MOUNTS

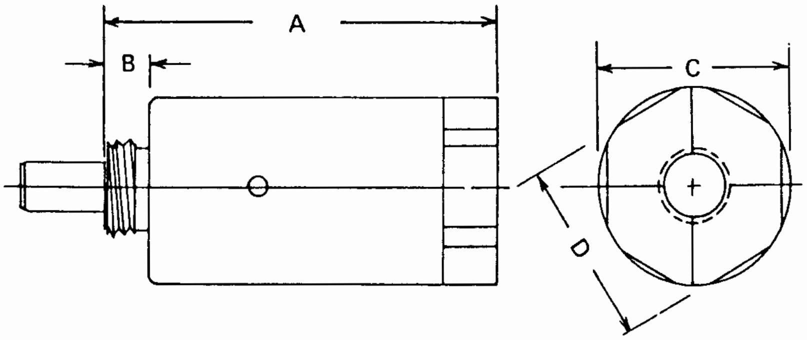
Specifications	Part Number	
	421-980-008	421-980-077
Stroke	1"	1-3/4"
Dimension A	2-29/32"	3-21/32"
Spring Load		
@ "O" Stroke	4 Lbs.	4 Lbs.
@ Max. Stroke	8 Lbs.	11 Lbs.
Equiv. Bore Dia.	1.5"	1.5"
Max. Oper. Pressure	125 PSI	125 PSI
Effective Area	1.7 in.^{2}	1.7 in.^{2}
Mounting Hole Sizes	1/4"-20 UNC	1/4"-20 UNC

Materials: Chrome-plated carbon steel rod. Die-cast aluminum cylinder and end cap. Oil-impregnated bronze bearing. Polyester fabric reinforced Nitrile diaphragm. Music wire spring.

Small Bore Cylinders



Now OEM's have the friction-free low hysteresis Air Cylinder they've dreamed of. Bellofram has applied its years of experience in manufacturing large sized friction-free Air Cylinders to provide OEM's with the first small bore diaphragm Air Cylinder available anywhere. A unique fabric reinforced rolling diaphragm guaranteeing friction-free seals has made the cylinder of your dreams come true. Available in selected standard sizes, with threaded or unthreaded rods, flush or extended; single-acting only, with spring return.



SPECIFICATIONS	PART NUMBER			
	311-908-013	311-908-034	311-908-014	311-908-035
A	2.81"	2.81"	1.95"	1.95"
B	0.438"	0.438"	0.244"	0.244"
C	15/16"	15/16"	15/16"	15/16"
D	7/8"	7/8"	7/8"	7/8"
Stroke	0.70"	0.70"	0.32"	0.32"
Spring Load				
@ "O" Stroke	2 lbs.	2 lbs.	5 lbs.	5 lbs.
@ Max. Stroke	7 lbs.	7 lbs.	7 lbs.	7 lbs.
Equiv. Bore Dia.	0.7"	0.7"	0.7"	0.7"
Max. Op. Pressure	125 PSI	125 PSI	125 PSI	125 PSI
Eff. Pressure Area	0.384 in.^{2}	0.384 in.^{2}	0.384 in.^{2}	0.384 in.^{2}
Nose Mount				
Thread Size	1/2"-20 UNF	1/2"-20 UNF	1/2"-20 UNF	1/2"-20 UNF
Rod Dia.	1/4"	1/4"	1/4"	1/4"
Rod				
Extension	Flush	3/4"	Flush	3/4"
Thread	–	1/4"-28 UNF	–	1/4"-28 UNF
Pipe Conn.	1/8"-27 NPSF	1/8"-27 NPSF	1/8"-27 NPSF	1/8"-27 NPSF

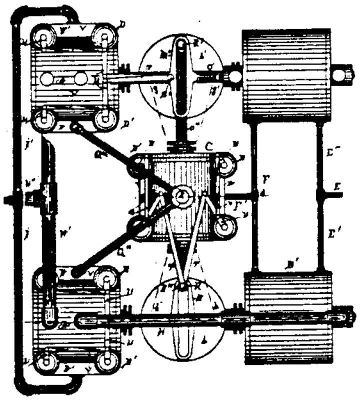
Materials: Carbon steel rod, aluminum alloy cylinder and end caps. Polyester fabric reinforced Nitrile diaphragm. Music wire spring. Oil-impregnated bronze bearing. Negligible breakaway friction.

U. S. PATENT OFFICE.

OFFICIAL GAZETTE.

AUGUST 30, 1904.

768,691. AIR-ENGINE. WILSON R. PRATT, Topeka, Kans. Filed Dec. 28, 1901. Serial No. 87,603. (No model.)



Claim.—1. The combination of straight-line engines, jackets provided on said straight-line engines, air-compressors communicating with said jackets, said compressors delivering to said straight-line engines, jackets provided on said air-compressors, an oscillating primary engine adapted to exhaust into said straight-line engines, and receiving air from said last-named jackets, substantially as described.

2. The combination with an oscillating primary engine, and its piston-rod, of rotating crank-disks, crank-pins on said disks, said piston-rod being connected to the said pins, double-bow stirrups in which the said crank-pins are arranged, straight-line engines, studs arranged in said bow-stirrups and connected with the said straight-line engines, and an air-compressor connected with said stirrups, substantially as described.

3. The combination with a high-pressure oscillating air-engine, of low-pressure straight-line air-engines receiving the exhaust therefrom, compressors driven by said engines, jackets for the compressors and straight-line engines, means whereby the compressors deliver to the straight-line-engine jackets, and means for conducting the air for the oscillating engine through the compressor-jackets, substantially as described.

4. In a device of the character described, the combination of an oscillating engine, straight-line engines adapted to receive the exhaust therefrom, means for compressing air adapted to be actuated by said engines, said means provided with air-jackets, and means for conducting the air for said oscillating engine through the jacket of said compressing means, substantially as described.

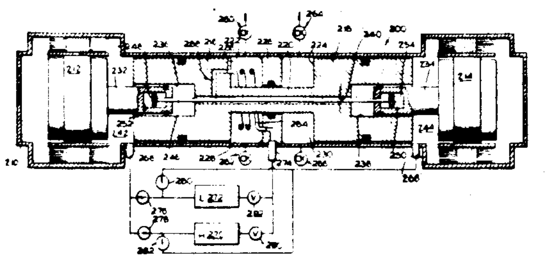
5. The combination with a primary high-pressure oscillating air-engine, of low-pressure straight-line engines receiving the exhaust therefrom, air-compressors actuated by said engines, and means whereby the compressed air by which the primary engine is actuated, is raised in temperature by the heat generated in the air-compressors, substantially as described.

6. The combination with a primary high-pressure oscillating air-engine, of low-pressure straight-line engines receiving the exhaust therefrom, air-compressors actuated by the said engines, semi-rotating valves arranged on the high and low pressure engines, means connected with said engines for operating said valves and means whereby the air by which the primary engine is actuated is caused to absorb heat generated in the compressors, substantially as described.

7. The combination with a high-pressure primary oscillating air-engine, of low-pressure straight-line engines receiving exhaust therefrom, jackets formed upon said last-named engines, compressors driven by said engines, jackets formed upon said compressors, means connecting said compressors with said jackets of the straight-line engines whereby the said compressors deliver thereto, means whereby the air for the oscillating engine passes through the said compressor-jackets, and means connecting said primary engine with the said straight-line engines and compressor-jackets.

JANUARY 5, 1971

3,552,120
STIRLING CYCLE TYPE THERMAL DEVICE
William T. Beale, Athens, Ohio, assignor to Research Corporation, New York, N.Y., a nonprofit corporation of New York
Feb. 1, 1967. This application Mar. 5, 1969, Ser. No. 812,530
Int. Cl. F03g *7/06;* **F25b** *9/00*
U.S. Cl. 60—24 **10 Claims**



A Stirling cycle thermal engine or refrigerating device wherein there is no primary mechanical connection between the displacer pistons and their associated power pistons, including various mechanical and pressure fluid means for varying the power output from or the power input to the device.

MACHINE DESIGN JUNE 21, 1984

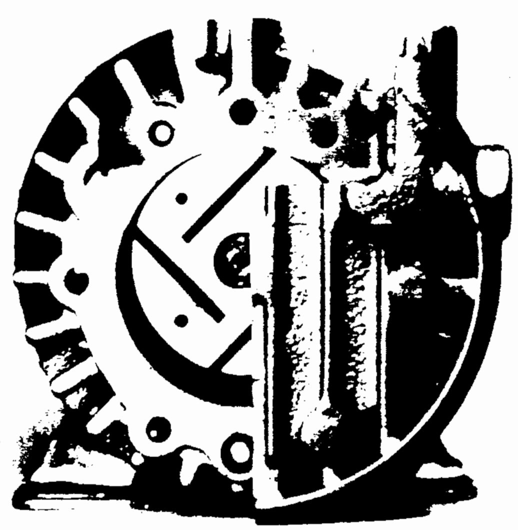
PNEUMATICS KICKS THE OIL HABIT

Once limited to low-pressure, low-flow applications, oil-less pneumatic systems are approaching the capabilities of lubricated systems.

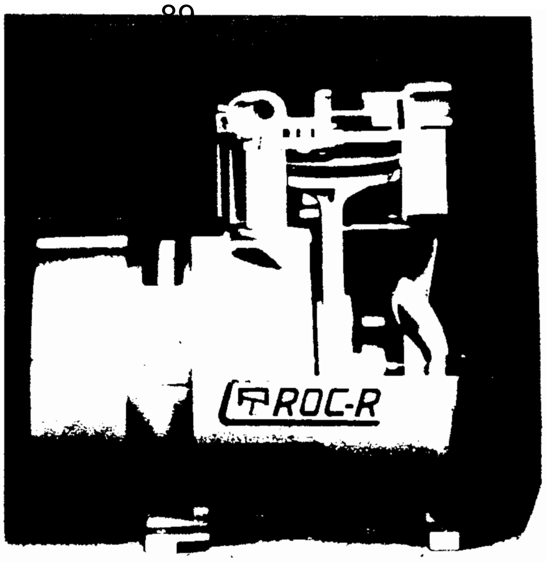
RICHARD C. BEERCHECK
Staff Editor

Oil is one of the biggest sources of contamination in a pneumatic system. It can damage the finished product, gum up components, and pose a hazard to personnel. To overcome these problems, manufacturers have developed a wide range of components that operate reliably without lubrication.

Three factors have spurred the move to oil-less pneumatic systems: the rising cost of maintenance and downtime, more stringent processing requirements for some products, and stricter government regulations on the amount of oil permitted in exhaust air.



Vane compressors were the first type converted to oil-less operation. The carbon vanes in this Gast Mfg. Corp. compressor are self-lubricating on cast iron in the presence of the water vapor commonly found in air.



Rocking piston compressors are a cross between reciprocating piston and diaphragm units. A self-lubricating PTFE cup on the piston of this Gast Mfg. Corp. compressor seals the compression chamber and reduces friction.

A typical plant compressed-air system can contain thousands of air-line lubricators. If each uses 8 oz of oil per week, the plant would need several people whose sole job is to fill the lubricators. No plant can afford this type of investment, and in actual practice most lubricators are serviced erratically. As a result, lubricators frequently run dry, leading to damaged valves, cylinders, and tools. To eliminate this cost and potential source of failure, many facilities are now demanding equipment that operates on oil-less air.

The second factor boosting the increased use of oil-less air systems is that many processes cannot tolerate even minute contamination of the finished product. Electronic-chip making, food processing, paper making, temperature controls, and textile manufacturing are typical processes that must avoid oil particulates in the exhaust air of pneumatic components.

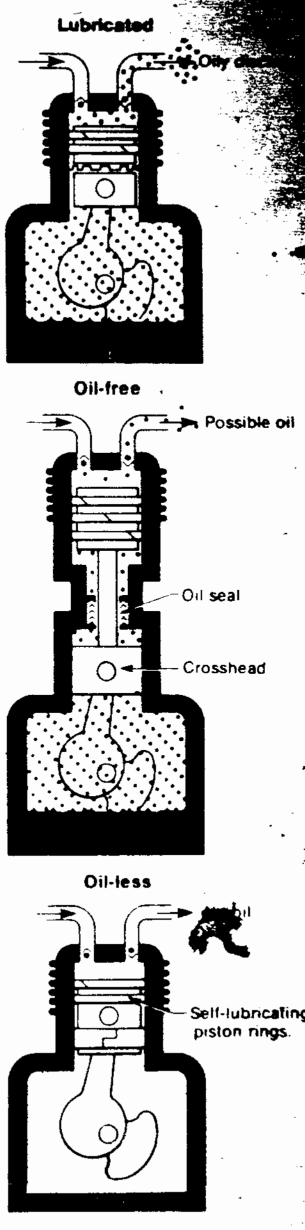
Finally, OSHA now strictly limits the amount of oil mist permitted in the factory environment. And breathing air from medical equipment or safety suits must be free of oil to prevent injury or illness.

Getting the oil out

Reciprocating compressors are the most widely used source of compressed air for systems operating below 25 hp. Normally, compressors operating over about 5 hp must be lubricated to ensure long life and efficient operation. Lubricated compressors have oil-filled crankcases; thus, oil vapor is always present in the compression chamber and discharge line. Filtration to remove oil from the air line requires constant maintenance. Besides, no filters are 100% effective at removing oil.

The first attempt to eliminate oily discharge from reciprocating compressors were the so-called oil-free compressors. These units contain a crosshead and oil seals to isolate the oil-filled crankcase from the compression chamber. The best seals ensure 99.99% oil-free air; however, even these seals wear eventually, increasing the possibility of oil in the air.

Oil-less compressors, as their name implies, use no oil at all. They feature self-lubricating piston rings and skirts, as well as sealed bearings. With no oil in the crankcase, none can be entrained in the discharge air. Problems with heat generation presently limit the horsepower capacity of oil-less compressors to about 15 hp.



Oil-less compressors

The heart of a compressed-air system is, of course, the compressor. Regardless of the type of compressor, the first design change required to make it oil-less is to use sealed bearings on the crankshaft and for piston compressors the wrist pin. Such bearings eliminate the need for splash or mist lubrication in the crankcase, the major source of oil particulates in a pneumatic system.

The main problems to overcome in switching from lubricated to oil-less operation are friction, heat, and higher noise levels. Friction has been reduced effectively through the use of self-lubricating materials. However, the temperature problem has limited the capabilities of oil-less compressors to about 15 hp, although higher-power units are on the horizon. The power limitation results because present-day materials cannot withstand the surface speeds and heat of higher-power operation. Noise problems are reduced by mounting the compressors in non-sensitive areas or by using readily available sound attenuating materials.

Besides power, the other major limitations of oil-less compressors is life. Presently, the sealed, grease-lubricated crankshaft bearings are rated for 8,000 to 12,000 h. Used in a typical 50% duty-cycle installation, these bearings provide about three years of operation before service is necessary. Similarly, the PTFE seals typically used in oil-less compressors are rated for 8,000 to 10,000 h of life.

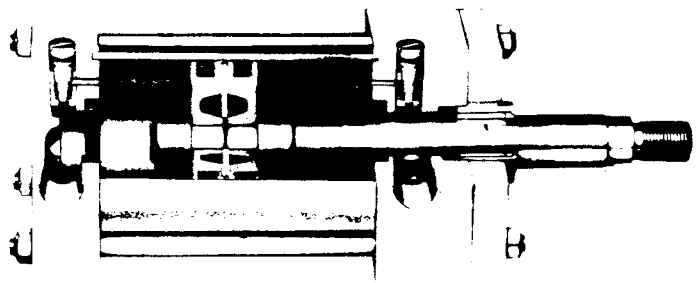
One of the first oil-less compressors was a vane compressor in which the phenolic vanes were replaced by carbon vanes. With the humidity commonly present in atmospheric air, carbon vanes are self-lubricating on cast iron. Over the years, the composition of the vanes has been improved to increase life; however, oil-less vane compressors are limited to a maximum operating pressure of about 15 psi and a maximum power of about 5 hp.

Diaphragm compressors, of course, have always provided oil-less operation because the diaphragm completely isolates the crankcase from the compression chamber. These compressors can produce up to 100 psi; however, their limited stroke restricts them to low-flow, low-power installations.

Oil-less piston compressors were developed in the mid-1950s. At first, these compressors used split carbon rings to seal the cylinder bore. However, such rings wore after only a few hours of operation, widening the ring gap and reducing flow capacity.

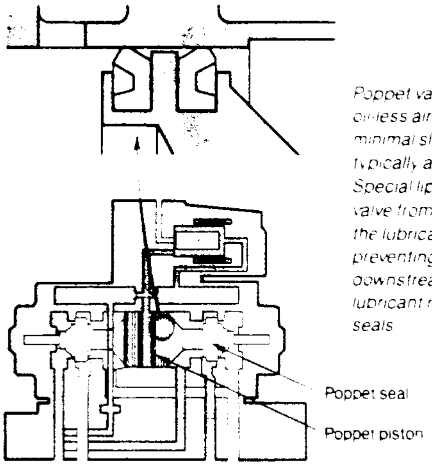
The problem of wearing rings has been overcome in most cases through the use of PTFE piston rings and skirts. PTFE rings effectively seal the cylinder bore and reduce friction. However, while such compressors can provide up to 100 psi, power is still limited.

The factor limiting the capacity of piston compressors is the

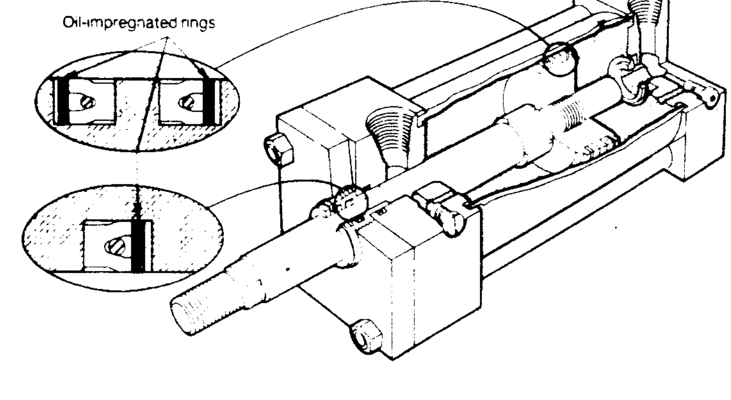


Oil reservoir stores lubricant in the piston of this cylinder from Lehigh Fluid Power Inc. The oil is wicked to the cylinder wall and is retained there by lip seals. An oil-impregnated bushing lubricates the piston rod.

Friction and heat are the biggest problems limiting the capacity of oil-less compressors. This compressor from ITT Pneumotive reduces friction and heat generation with force-compensated PTFE piston rings and skirts. An integral blower and shroud (not shown) direct cooling air over the finned cylinders and crankcase.



Poppet valves are widely used in oil-less air systems because they have minimal sliding contact. Such valves typically are prelubricated at assembly. Special lip seals, such as those on this valve from Parker Hannifin Corp., retain the lubricant in the poppet piston area, preventing it from being exhausted downstream. Some valves include a lubricant reservoir between the lip seals.



Self-lubricating cylinder from Parker Hannifin Corp. has oil-impregnated rings in the piston and rod gland. The rings dispense oil to the cylinder wall and rod. Lip seals retain the oil in these areas and prevent it from being exhausted to the atmosphere.

heat produced in the compression process, which reduces the life of the bearings and seals. To boost capacity, manufacturers of piston compressors have devised innovative ways to reduce heat buildup and remove the heat that does develop.

The usual way to reduce heat generation is with so-called force-compensated piston rings. The rings ride on a cushion of air, sealing during the compression stroke but releasing on the intake stroke. This mode of operation reduces the friction and pressure forces on the rings, cutting heat generation and improving wear life. Any heat that is generated is removed by forced-air cooling. A blower wheel mounted on the end of the motor shaft directs cooling air through the crankcase and over the pistons, cylinders, and bearings, permitting the compressors to operate continuously at up to 15 hp and pressures to 200 psi.

A third type of [illegible] rocating compressor, [illegible] piston type, is a cross [illegible] piston and diaphragm [illegible] pressor. The piston is sealed [illegible] PTFE cup that expands [illegible] forced into the cylinder. The expansion compensates for the rocking motion of the piston and maintains a tight seal on the cylinder wall. Rocking-piston compressors can produce up to 100 psi, but maximum power rating is under 1 hp.

Nonlubricated valves

The designation "nonlubricated" for an air valve usually means that the valve is prelubricated or contains parts made of a self-lubricating material. Valves are prelubricated in one of two ways: 1. The valve parts are dipped in a lubricant prior to assembly. 2. The valve includes a reservoir of lubricant that is fed to critical areas. In either case, the valves normally do not require further lubrication in service.

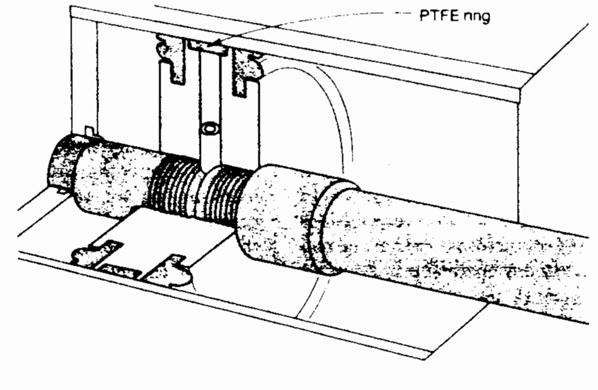
The design of pneumatic directional-control valves to operate without air-line lubrication requires special considerations. And there is considerable debate over the types of valves suitable for oil-less service. For instance, some experts contend that spool valves, with sliding action between parts, can experience problems with wear and high shifting forces when the lubricant either wears off or is carried away by the air stream. Also, past experience with packed-spool valves has shown that the synthetic rubber seals commonly used in such valves can actually wear away the metallic sealing surfaces if the valves are not lubricated.

However, some manufacturers of packed-spool valves replace the rubber seals with self-lubricating PTFE seals, and they hardcoat the valve bodies. This combination provides a low-friction, high-lubricity sliding contact that supporters claim provides reliable operation on nonlubricated air.

In general, lapped-spool and poppet valves are the most widely used types for oil-less service. Lapped-spool valves depend on the close fit between body and spool to seal the valve. The valves typically are dipped in a lubricant prior to assembly and require no further lubrication in service.

However, lapped-spool valves can have problems when operating on nonlubricated air, because they are susceptible to clogging from contaminants or from hydrocarbons carried downstream from a lubricated compressor. Hydrocarbons are particularly troublesome, because the heat of compression transforms them into varnishes that can build up in the valve. When the valve sits idle for long periods, such as over a weekend, the varnish can cause the spool to bind, preventing proper shifting and resulting in solenoid burnout. Manufacturers claim that lapped-spool valves are easily protected with coalescing filters. Such filters effectively remove hydrocarbon aerosols and solid particles from the air.

In contrast to other valve



PTFE ring on the piston of this Mosier Industries Inc cylinder lubricates the piston cylinder-wall contact The cylinder wall is highly polished to reduce friction, and an oil-impregnated bushing lubricates the piston rod

types, poppet valves have no sliding action in the primary sealing area. Thus, the valves resist clogging from air-line varnish and contaminants. The only area of a poppet valve with sliding contact is the piston or actuation section. But this area can be prelubricated to operate for millions of cycles through the use of either self-compensating seals or an internal lubricant pocket.

Valves designed for normal service usually are prelubricated and have self-compensating seals. In such valves, lip seals on the piston are specially designed to hold the lubrication in place, and they wipe the bore clean with each stroke. Because the seal lips are self-compensating, sealing is maintained even as components wear during operation. Such valves also can have bodies made of self-lubricating injection-molded plastic, further reducing the need for lubrication.

Valves for heavier-duty service usually include a lubricant reservoir between the piston lip seals. The lubricant is fed continuously to the piston/cylinder-wall interface and is retained in this area by the lip seals. In tests, such valves have survived over 100 million cycles on nonlubricated air.

Lubed-for-life cylinders

Over the years, there has been much misunderstanding about the true meaning of nonlubricated cylinder requirements. Basically, nonlubricated cylinders fall into three categories: prelubricated, unlubricated, and self-lubricating. Self-lubricating cylinders can be further categorized as those with lubricant reservoirs and those with self-lubricating parts.

Prelubricated cylinders are the oldest type of nonlubricated cylinder. At assembly, an extreme-pressure grease is applied to the seals, piston, cylinder bore, and gland surfaces. The lubricant adheres tightly to the cylinder parts, resisting explusion into the atmosphere. Tests of such cylinders have shown that the moisture commonly present in compressed air also helps to lubricate the piston seals.

Unlubricated cylinders are constructed of absolutely dry parts with no lubricant. Requirements for such cylinders arose in the food industry because of concerns about the possibility of contamination. However, unlubricated cylinders require special, expensive seals, and they have had only limited success in tests. Further evaluation of the possible use of unlubricated cylinders has shown that a truly unlubricated design is not absolutely necessary. As a result, the demand for this type of cylinder has never really developed.

Self-lubricating cylinders were developed to meet an auto-industry specification calling for cylinders to operate 20 million cycles on dry, moisture-free air. To survive such severe service, these cylinders incorporate either an internal lubricant reservoir or self-lubricating piston-ring and rod-seal materials.

There are several different approaches to providing a lubricant reservoir. In some cylinders, special composite rings impregnated with extreme-pressure oil are used in the piston and rod-seal areas. The rings are positioned to dispense oil to the cylinder bore and piston rod during each stroke. Special lip seals collect excess lubricant so that none escapes into the atmosphere.

Other cylinders incorporate an oil reservoir in the piston, which is filled at assembly. The lubricant is then wicked to the cylinder wall. Again, lip seals collect excess lubricant, preventing it from escaping to the atmosphere.

Cylinders with self-lubricating parts commonly include a phenolic or PTFE wear ring on the piston and either an oil-impregnated bushing or wear ring in the rod gland. The cylinder bore and rod also are highly polished to reduce friction. These cylinders can be either prelubricated or unlubricated at assembly.

Technical assistance provided by Gast Mfg. Corp., Benton Harbor, MI, Humphrey Products, Kalamazoo, MI ITT Pneumotive, Monroe, LA; Parker Hannifin Corp., Pneumatic Div., Otsego, MI, Ross Operating Valve Co., Detroit, MI, Finite Filter Co. Inc., Oxford, MI, Lehigh Fluid Power Inc., Lambertville, NJ

Filtration — key to oil-less operation

Because of the power limitations of oil-less compressors, most large air systems continue to use lubricated compressors. Estimates of the hydrocarbon content in the discharge air of such compressors are:

- Screw — 25 to 75 ppm at 200°F.
- Reciprocating — 5 to 50 ppm at 350°F.
- Centrifugal — 5 to 15 ppm at 300°F.

At a concentration of only 25 ppm, a typical compressor flowing 100 scfm for only 35 h will dump 8 oz of oil into the pneumatic system. To prevent the oil from contaminating components or the finished product, a coalescing filter is essential.

Coalescing is a continuous air filtration process in which liquid aerosols contact glass microfibers and are agglomerated into larger and larger droplets. The droplets eventually emerge on the downstream side of the filter and are drained away.

The coalescing process occurs in three ways, depending on aerosol size and inertial force. Aerosols larger than 2 μm are collected by direct impact, with larger droplets being formed by successive impingements. Aerosols from 0.2 to 2 μm — the predominant size range — are collected by interception by the glass fibers. Aerosols in the 0.001 to 0.2 μm size range follow random Brownian motion, striking the fibers through diffusion and being held by van der Waals forces.

The standard coalesing filter removes 99.97% of the aerosols in the 0.3 to 0.6 μm range, is 99.98% efficient at removing all aerosols, and removes all solid particles larger than 0.3 μm. Thus, incoming air at a 20 ppm level is reduced to 0.004 ppm concentration, which is acceptable for almost any critical pneumatic system. More efficient filters rated at over 99.9999% aerosol removal also are available.

Coalescing filter is made of borosilicate glass microfibers vacuum formed into interlocking contact. This filter, from Finite Filter Co. Inc., has a rigid porous support tube and a glass matrix drain layer. It is compatible with all compressor lubricants, and removes oil aerosols and solid particles from the air stream

TORQUE CONVERTERS / CHAPTER 6

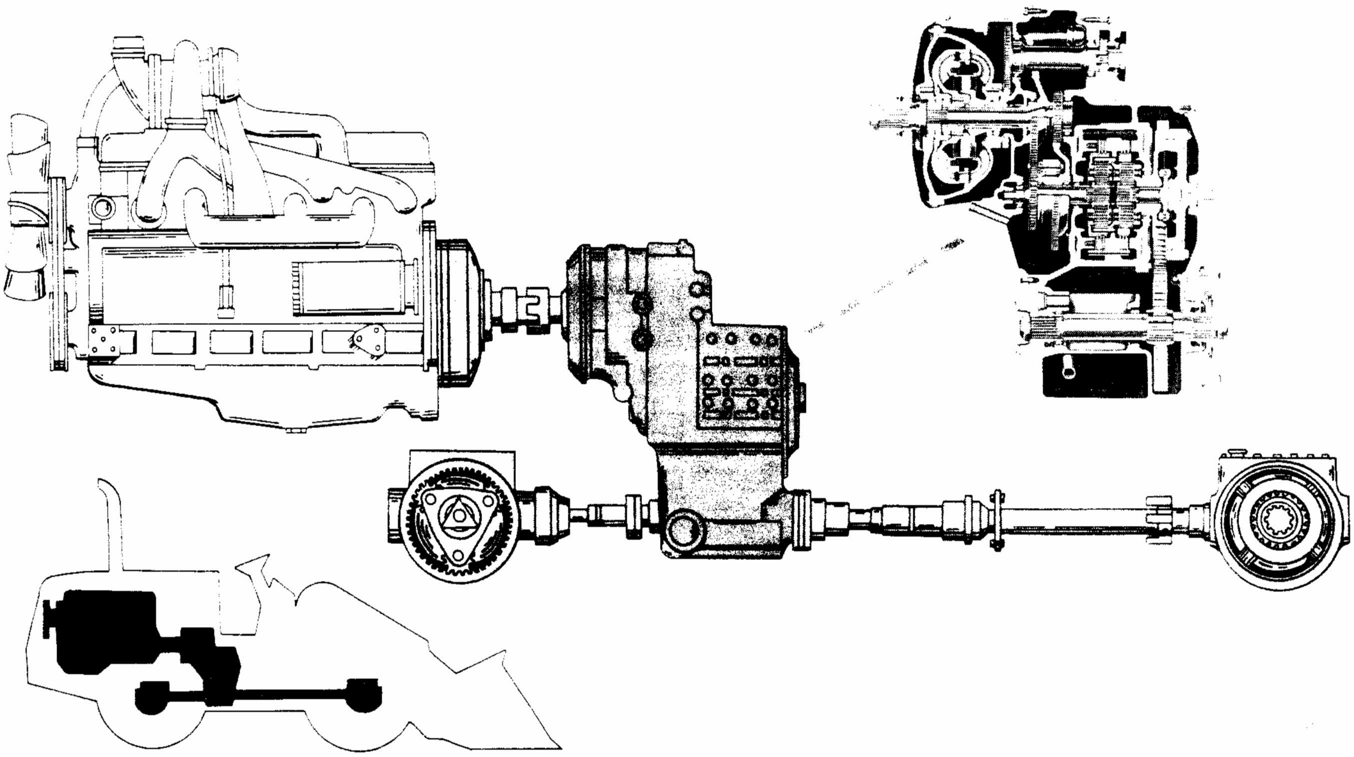


Fig. 1—Torque Converter In Complete Power Train

INTRODUCTION

A **torque converter** is an automatic *fluid* drive. It transmits engine torque by means of hydraulic force, shifting smoothly through an infinite number of speeds.

The automatic transmission of an automobile automatically shifts gears in response to torque requirements in addition to the automatic response of the torque converter which is a part of the automobile's automatic transmission system.

Actually a gear train is used with the torque converter to give extra speed ranges. But no gear train could give the infinite variations in speed and torque of a torque converter.

Acting as a *clutch*, the torque converter connects and disconnects power between the engine and the gear train. As a *transmission*, the converter gives many more speed ratios than are practical with a strictly mechanical gear box.

To compare a torque converter with a hydrostatic drive (Chapter 5), use this rule of thumb:

Hydrostatic drives are driven by fluids at *high pressure* but at relatively *low velocity*, while **torque converters** are driven by fluids at *low pressure* but at *high velocity*.

Here are the formulas:

- **Hydrostatic Drive** = *HIGH pressure* - *LOW velocity*
- **Torque Converter** = *LOW pressure* - *HIGH velocity*

HOW IT WORKS

To understand a torque converter, we must first look at a basic **fluid coupling.**

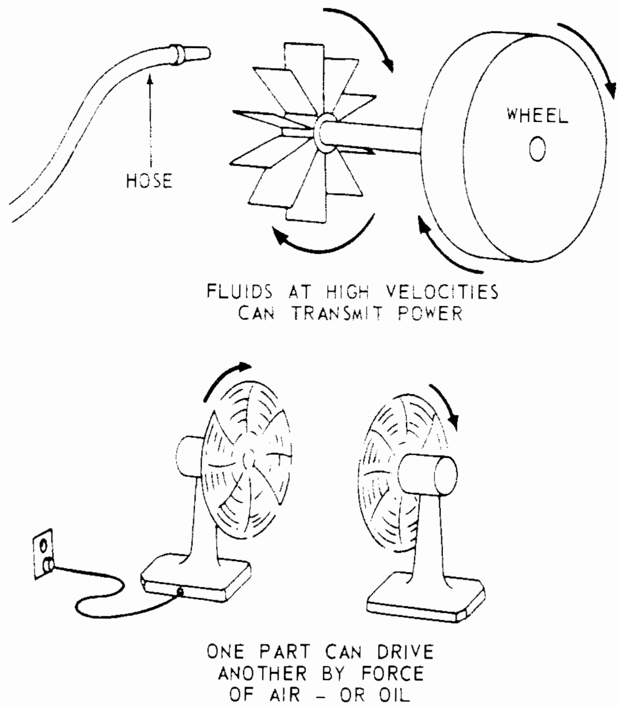


Fig. 2—Basic Principles Of A Fluid Coupling

The basic principles of all fluid couplings are shown in Fig. 2.

At the top, a fluid at high velocity strikes a turbine and forces it to turn, driving the wheel. Thus, **torque is transmitted by a fluid.**

To change this torque, the velocity of the fluid is changed. At low velocity, the fluid will not even move the turbine. At higher velocity, the turbine starts turning and the wheel picks up speed.

This is something like putting two electric fans face to face as shown in the lower part of Fig. 2. By plugging in only one fan, we can cause the other one to rotate.

This principle is used in a fluid coupling as follows:

Inside an oil-filled housing (A, (Fig. 3) are two parts: the driving half, or **pump** (impeller), and the driven half, or **turbine**.

As the pump is turned by the engine, centrifugal force causes oil to be forced radially outward, crossing over and striking the vanes of the turbine. This rotates the turbine in the same direction and so couples the power.

Drawings B, C, and D explain how the flow of oil drives the turbine.

In B, fluid is placed in a bowl and lies level.

In C, the bowl is spun rapidly and centrifugal force causes the fluid to climb up and spill over the outside edge of the bowl.

In D, another bowl is placed down over the first one. Now when the bowls are spun an axial flow or circuit is created and turning force is transmitted between the driving bowl and the driven bowl.

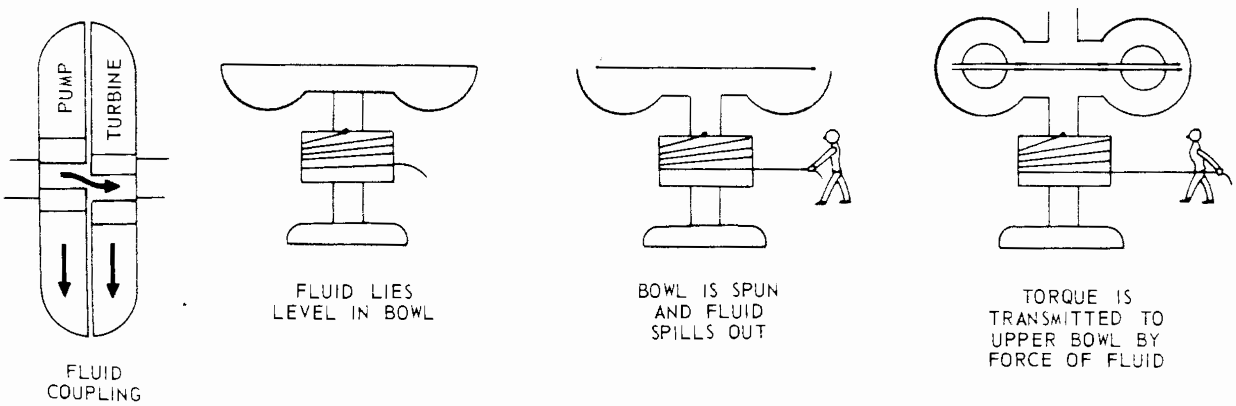


Fig. 3—Operation Of A Fluid Coupling

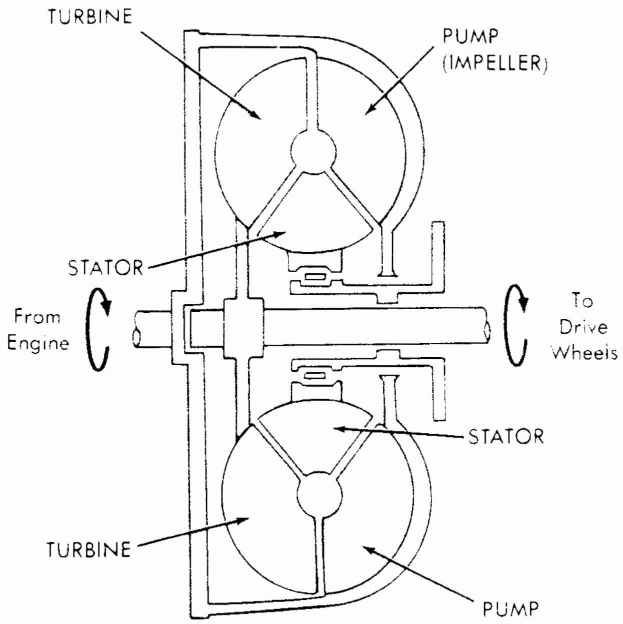


Fig. 4—Torque Converter

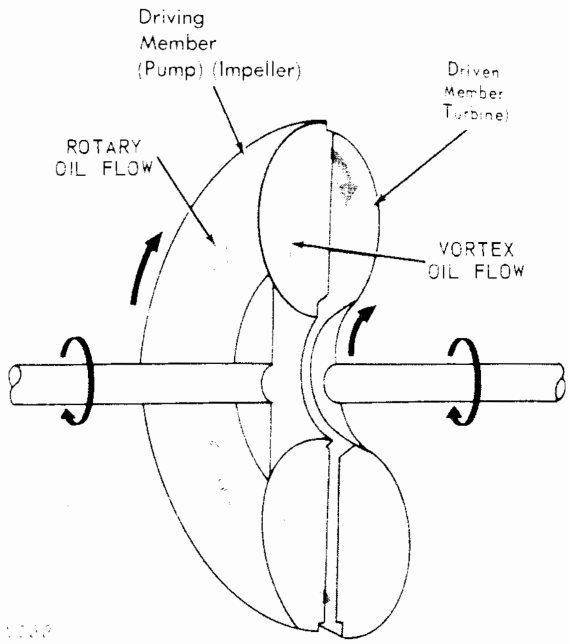


Fig. 5—Vortex And Rotary Flow In A Fluid Coupling

Torque is thus transmitted, but it is not increased.

This is where the **torque converter** goes beyond the basic fluid coupling, for the converter can **multiply torque.**

A torque converter (Fig. 4) looks much like the fluid coupling we have just described. The main difference is that the torque converter has—in addition to the driving pump and the driven turbine—a set of blades or vanes called a **stator.**

The stator vanes change the direction of oil flow after it has gone through the turbine and sends it back to the pump. This enables the pump to increase the twisting force or *multiply* the torque.

Since the converter is a closed unit, this flow is repeated continuously. Many streams of fluid act against many vanes at once and this is what gives the power to drive a heavy machine.

OIL FLOW IN CONVERTER

Let's look at the flow of oil in the converter during two cycles:

1) Increasing the torque

2) Decreasing the torque

INCREASING THE TORQUE

Remember that the pump is driven by the engine, while the turbine receives fluid energy from the pump and sends it to the drive wheels.

Also remember how centrifugal force sets up a continuous circular flow in the coupling (Fig. 5).

This circular flow of oil *between* the pump and turbine is called **vortex flow.**

Another flow is set up *around* the pump and turbine to form a coupling: this is called a **rotary flow.**

The action of these combined oil flows will transmit torque *but not increase it.*

Increasing the torque is where the stator comes into play.

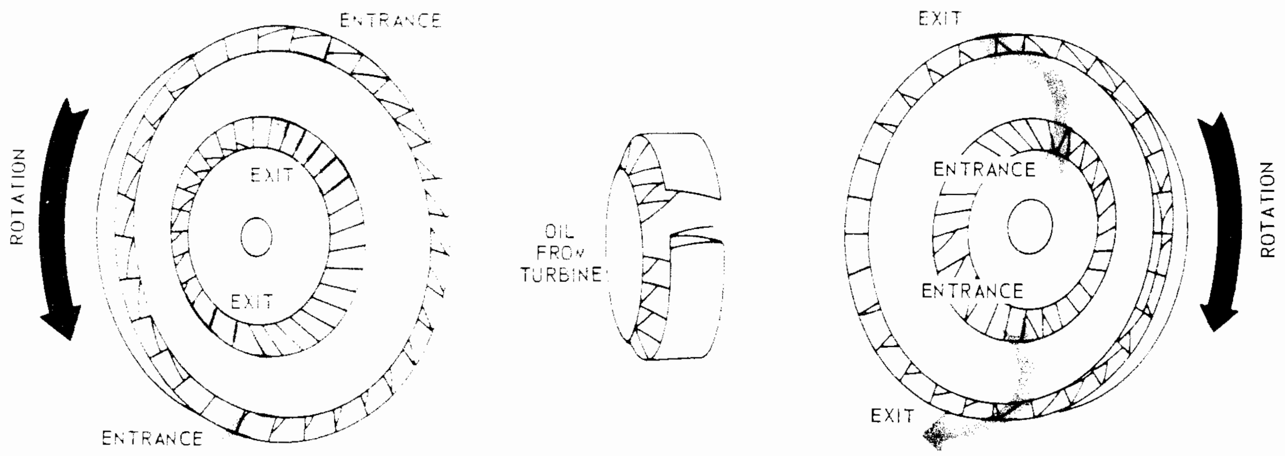


Fig. 6—Oil Flow Through Pump, Turbine, And Stator

Fig. 6 shows how oil emerges from the turbine in reverse compared to pump rotation. Unless this oil flow is turned around, it will cause a loss of power.

Note in Fig. 6 that the oil passages at the rim of the turbine where the oil enters must become smaller as they approach the smaller diameter of the turbine. As the same volume of oil must squeeze through these funnel-like passages, the oil stream will speed up when it leaves the turbine. This speed is used to increase torque by directing it against the **stator**, which acts as a fluid lever or fulcrum. The stator changes the direction of flow and sends the oil into the pump in the same direction as pump rotation.

Let's see how the stator does its job (Fig. 7). A stream of oil aimed at a flat surface (A) splashes off at various angles. The oil can be made to flow more smoothly by curving the entrance (B) and can be reversed by more curving (C) with a resulting increase in force as indicated by the large arrow.

The stator has curved blades (as in C) which the oil strikes as it leaves the turbine. These blades turn the oil back in the direction of pump rotation.

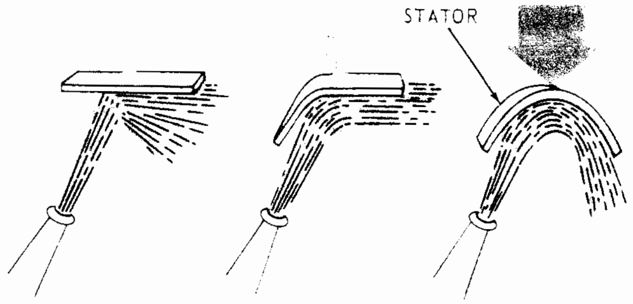


Fig. 7—Stator As A Fluid Lever

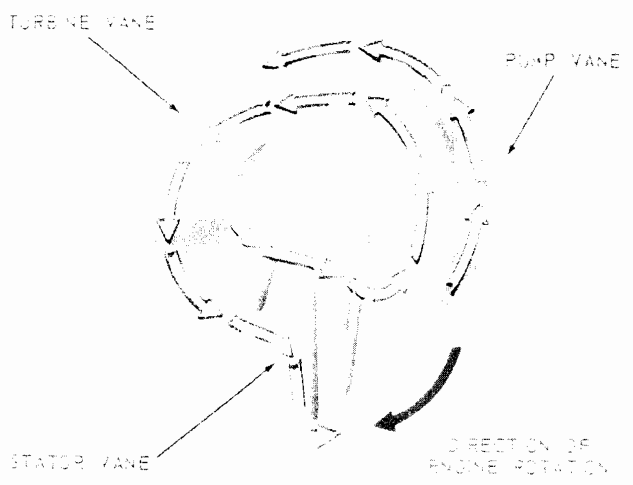


Fig. 8—Flow Direction Is Reversed By The Stator Vanes

Now that the stream of oil is moving in the same direction but at a greater velocity, it enters the pump smoothly (Fig. 8). Its velocity is added to that developed in the pump so that the total velocity at the pump exits has been increased.

This regenerating action is the key to multiplying torque in a torque converter.

To change the direction of oil flow, the stator must be stationary during the increase in torque. However, once the pump and turbine are turning at the same speed it would create resistance. Therefore, the stator is sometimes mounted on a freewheel clutch so that it can turn in one direction only (once torque stops increasing). (In other torque converters, the stator may be fixed to the converter case.)

DECREASING THE TORQUE

Torque is increased as long as the engine is accelerating to get the machine under way. But as the engine speeds up, the turbine also speeds up, which causes the vortex flow of oil in the converter to decrease. At the same time, the rotary flow increases.

Vortex flow keeps on changing to *rotary* flow (Fig. 9) until the pump and turbine are "locked up" and all torque increase stops.

The torque converter now acts as a simple fluid coupling, sending the same torque it receives on to the drive wheels.

The torque converter is able to automatically reduce or increase torque in infinitely small steps to match the needs of the machine and its driver. This has the same effect as shifting speeds in a gear transmission except that it is done smoothly and automatically while "on the go."

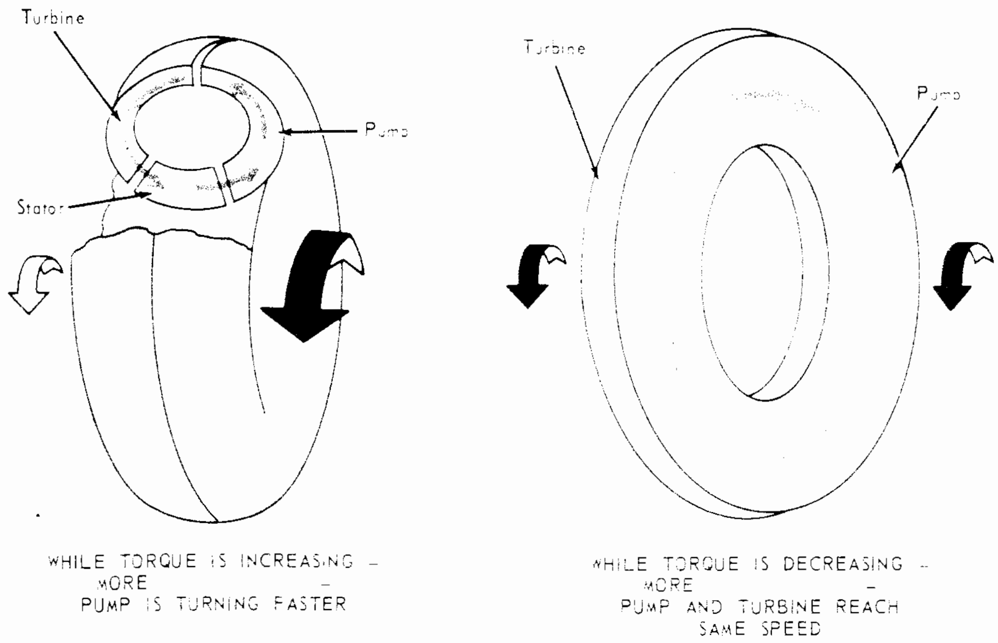


Fig. 9—How Vortex Flow Changes To Rotary Flow As Torque Is Reduced

VARIATIONS ON TORQUE CONVERTERS

The chart below shows some of the variation in the number of elements used in several torque converters.

Torque Converter Elements	*Design A*	*Design B*	*Design C*	*Design D*
Pump	*2*	*1*	*1*	*1*
Stator	*2*	*1*	*2*	*1*
Turbine	*1*	*2*	*1*	*1*

The design of a torque converter must match the engine torque and road speed for each application. In off-the-road machinery, units are matched to horsepowers (kilowatts) ranging from 40 to 600 (30-450). But the basic principles we've covered apply to all.

TORQUE CONVERTER TRANSMISSIONS

The torque converter is but one part of the complete transmission (Fig. 10).

Here are the major components:

- **Converter Section**
- **Range Gear Section**
- **Final Drive Section**
- **Hydraulic Control System**

Let's look at each part and see what is does for the complete transmission.

CONVERTER SECTION

We have looked at torque converters with three elements—one pump, one turbine, and one stator.

Now we will examine a *twin-turbine* model (Fig. 11) which has one pump and one stator, but *two* turbines (first and second).

The first turbine is shown in blue, while the second is shown in red.

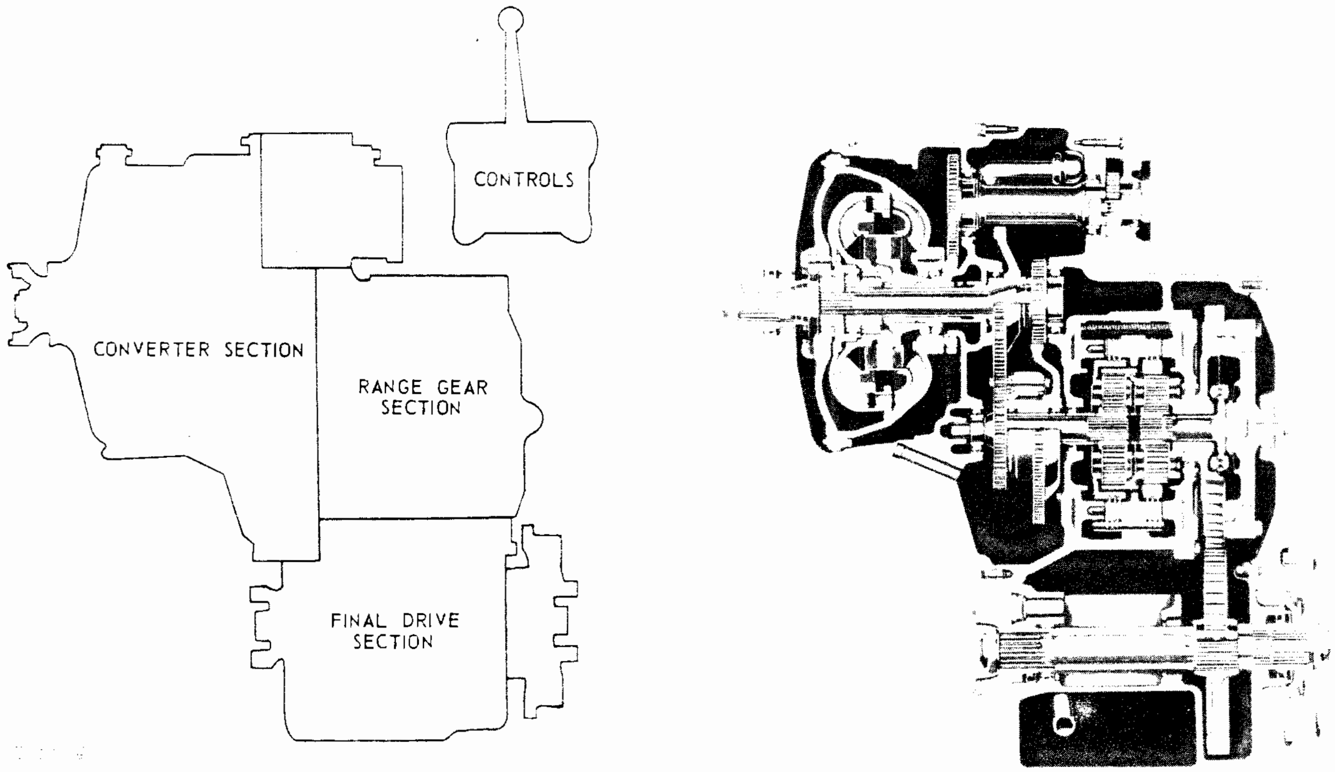


Fig. 10—Complete Torque Converter Transmission

Each turbine is connected to its individual output gear set. In reality, two converters have been combined into one. The first (blue) turbine is connected to its output shaft by a freewheel clutch.

Here is how the two turbines work together:

When torque demand is high, the freewheel clutch is engaged and the first turbine, assisted by the second turbine, drives the gears. When the machine speeds up, torque demand drops. Then the second turbine takes over the entire load and the freewheel clutch disengages the first turbine.

As a result, the first turbine provides *high torque and low speed*, (for starting up and loading) while the second turbine provides *higher speed with lower torque* (for travel).

A combining gear set directs torque from the first and second turbine (or the second turbine only) to the range gear section.

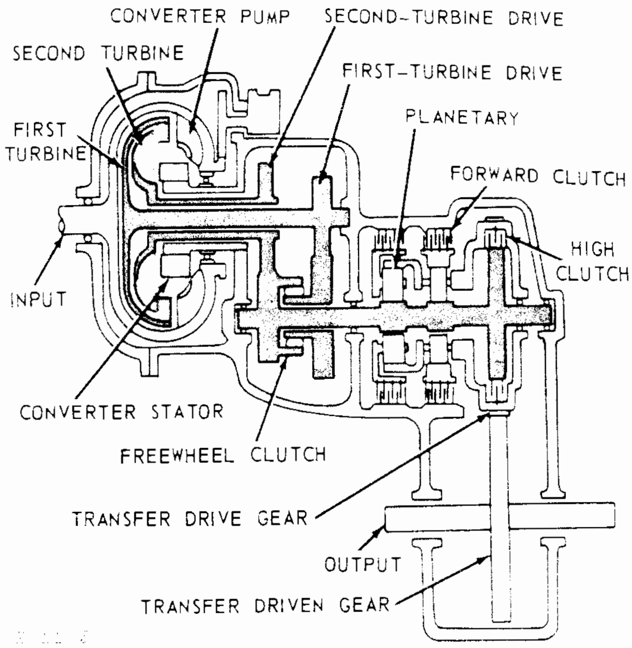


Fig. 11—Twin-Turbine Torque Converter

RANGE GEAR SECTION

Since torque is reduced and increased automatically in the torque converter, only a few gear sets are normally required in the transmission.

However, as the torque converter rotates in only one direction, it is necessary to have a reverse gear. In some applications, it is also desirable to have low and high range gears as shown in Fig. 11.

Simple planetary gear sets meet the needs of extra gear ranges and lends themselves to hydraulic control.

NOTE: Planetary gear sets and their control by hydraulic clutches are explained in Chapter 4.

FINAL DRIVE SECTION

The final drive section includes the transfer drive gear, transfer driven gear and output shaft (Fig. 11). The output shaft provides for one or for two outputs from the same common shaft. Two outputs can be used to propel a four-wheel drive machine as shown in Figs. 11 and 15.

Note also that by adding the transfer gear and one clutch (high range) in Fig. 11, another forward speed range can be obtained.

HYDRAULIC CONTROL SYSTEM

The hydraulic control system uses oil to do these jobs:

- **Oil flow lubricates and cools the parts**
- **Oil pressure engages the clutches**
- **Oil velocity drives the turbines**

Let's use the hydraulic controls for the twin-turbine converter we have just described to see how a typical system works.

There are four basic circuits as shown in Fig. 12:

1) Oil Pump and Filter Circuit (shown by blue lines).

2) Main-Pressure Regulator Valve and Converter-In Circuit (shown by red lines).

3) Converter-Out, Cooler, and Lubrication Circuit (shown by dotted blue lines).

4) Selector Control Valve Circuit (shown by dotted red lines).

Let's build up the system and explain each circuit.

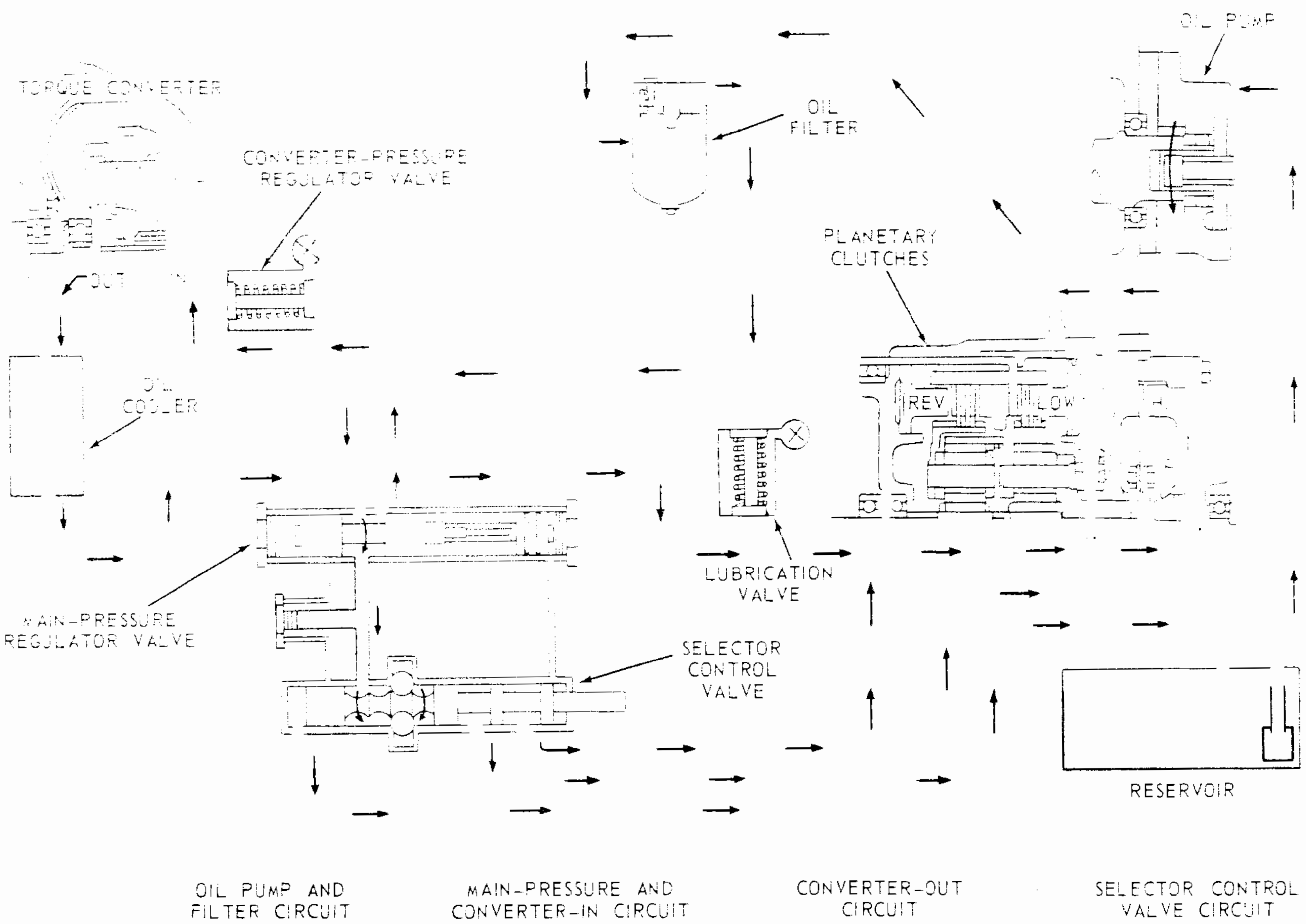


Fig. 12—Hydraulic Control System For Torque Converter Transmission

Oil Pump And Filter Circuit

Oil is drawn from the transmission reservoir by the oil pump as shown in Fig. 12. The pump delivers its entire output to a full-flow oil filter for cleaning. From the oil filter, the oil supply is sent to the main-pressure circuit.

Main Pressure Regulator Valve and Converter-In Circuit

The main pressure regulator valve provides pressure for the planetary clutch packs, directs oil to the selector control valve, and supplies pressure oil into the torque converter. A converter pressure regulator valve in the converter-in line limits the oil pressure there.

Converter-Out, Cooler, and Lubrication Circuit

The torque converter is continuously filled with oil during operation. Rotation of the converter pump imparts energy to the oil which, in turn, drives the turbines. The oil then flows between the stator vanes which redirect it to the pump.

Oil flowing out of the converter is directed into the oil cooler as shown in Fig. 12. The cooler is a heat exchanger in which the oil flows through water- or air-cooled passages.

From the cooler, oil flows to all passages and outlets in the lubrication circuit (dotted blue lines). A lubrication valve between the cooler and lubrication system returns all excess oil to the transmission reservoir.

Selector Valve Circuit

Pressure oil from the main-pressure regulator valve flows into the selector valve bore and surrounds the valve in the area of the detent notches. From this area, main pressure oil is available for operating the low, high, and reverse range planetary clutches (see dotted red lines).

Moving the selector valve allows oil to charge the selected clutch line and to engage that clutch.

This completes the four basic control circuits in our torque converter transmission.

NOTE: For details on hydraulic components, see the FOS "Hydraulics" manual.

SUMMARY: FEATURES OF TORQUE CONVERTERS

1. *Multiply torque*
2. *Provide infinite speed ranges*
3. *Shift smoothly and automatically*
4. *Cushion shock loads on drive lines*
5. *Help to dampen vibrations*

TROUBLESHOOTING

INTRODUCTION

Oil circulates at high velocity within the torque converter and any foreign material it carries will rapidly wear down the edges and pit the turbine vanes (Fig. 13), changing their effective shape.

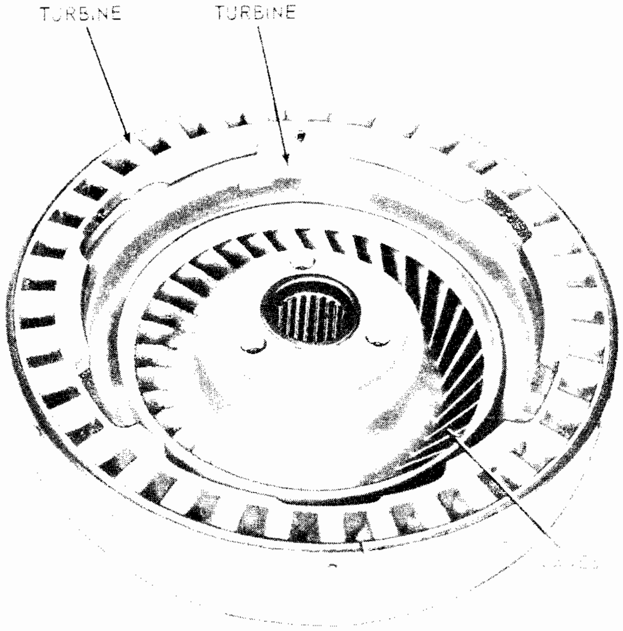


Fig. 13—Turbines For Twin-Turbine Torque Converter

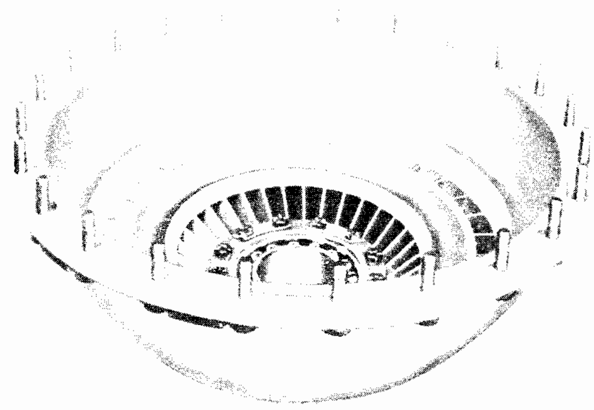


Fig. 14—Pump For Torque Converter

Vane damage will also cause the turbines to become unbalanced. In addition, dirty oil will damage bearings and seals.

Some torque converters contain parts made of lightweight aluminum alloys (Fig. 14). Converter housings are usually made of cast aluminum.

Be sure to handle all converter parts carefully to prevent nicking, scratching, and denting.

Parts which fit together closely but with operating clearance will stick if damaged only slightly. Parts which depend upon smooth surfaces for sealing may leak if scratched.

All these parts should be carefully handled and protected during removal, cleaning, inspection, and installation.

Use these rules to help prevent failures:

1. *Be sure the oil is kept clean.*
2. *Service the system at proper intervals.*
3. *Repair it only if you are qualified.*
4. *Use all special tools recommended.*
5. *See the machine Technical Manual for details.*

PRELIMINARY INSPECTION

The operator and serviceman can use his eyes, ears, and even his nose to head off trouble before serious damage is done. Here's how:

Look *Check oil levels.* Refer to the operator's manual.

Check gauges. When starting, running or stopping, watch the gauges: they tell the story in terms of pressure and temperature.

Inspect for leaks. Leaks, crimped oil lines, and clogged filters — all affect the converter output.

Examine the oil. Look for water, dirt, and particles from the converter and the clutch plates.

Listen *Unusual noises.* Listen for squealing from a stuck valve, or grinding or grating sounds from inside the converter.

Smell *Overheating.* A strong odor of overheated oil is a major trouble sign. Find the cause at once.

Because one trouble can have the same symptoms as another, only good instruments in the hands of trained servicemen can detect the difference.

Once trouble has been detected in the field, check out the unit with the proper testing tools.

TROUBLESHOOTING PROBLEMS

The troubleshooting given here will cover four common areas of trouble:

1. *Overheating*
2. *Noise*
3. *Leaks*
4. *Machine Response*

Let's examine each kind of trouble.

1. Overheating

Overheating is a major problem in converter operation. It is affected by the design, the type of work, the operator, the air temperature, and the condition of the unit.

Overheating can cause a loss of power and can damage seals and gaskets and warp metal parts.

A converter may overheat if the *work is heavy*— not always, but heavier work will generally mean more heat.

If the converter is *undersized* for the normal work of the machine, it will operate at low efficiency and will tend to overheat. The operator can usually relieve the load on the converter by operating in a lower gear, or by reducing the load.

Whenever you receive complaints that a converter in a machine is overheating, try to find out by watching or by questions whether the correct gears and work methods are being used.

Air in the converter will also cause overheating. Torque converters can work properly only if they are filled with fluid. Air mixed with the oil will cause poor performance, overheating, and possibly serious damage.

Air may enter the system:

(1) If the fluid in the reserve tank (if used) is low enough to permit the charging pump to suck air.

(2) If a moderately low level of oil permits sucking of air while the machine is working on steep slopes.

(3) If there is a leak in the suction line pump gaskets or O-rings (suction leaks may be too small to show up by outside leaking of oil).

(4) If the oil or filters are changed or when the lines are opened for any reason.

SUMMARY: OVERHEATING

Let's recap the major causes of overheating in the torque converter.

First, overheating is not only a major problem, it is also a major symptom.

Although normal operating temperatures can be exceeded very rapidly, a machine using a torque converter matched to the job should not exceed its normal temperature when properly used.

Second, the cause of overheating may be found in one or more areas outside of the converter.

Before deciding that the converter is the cause of overheating, check these possibilities:

(a) Air in fluid system.

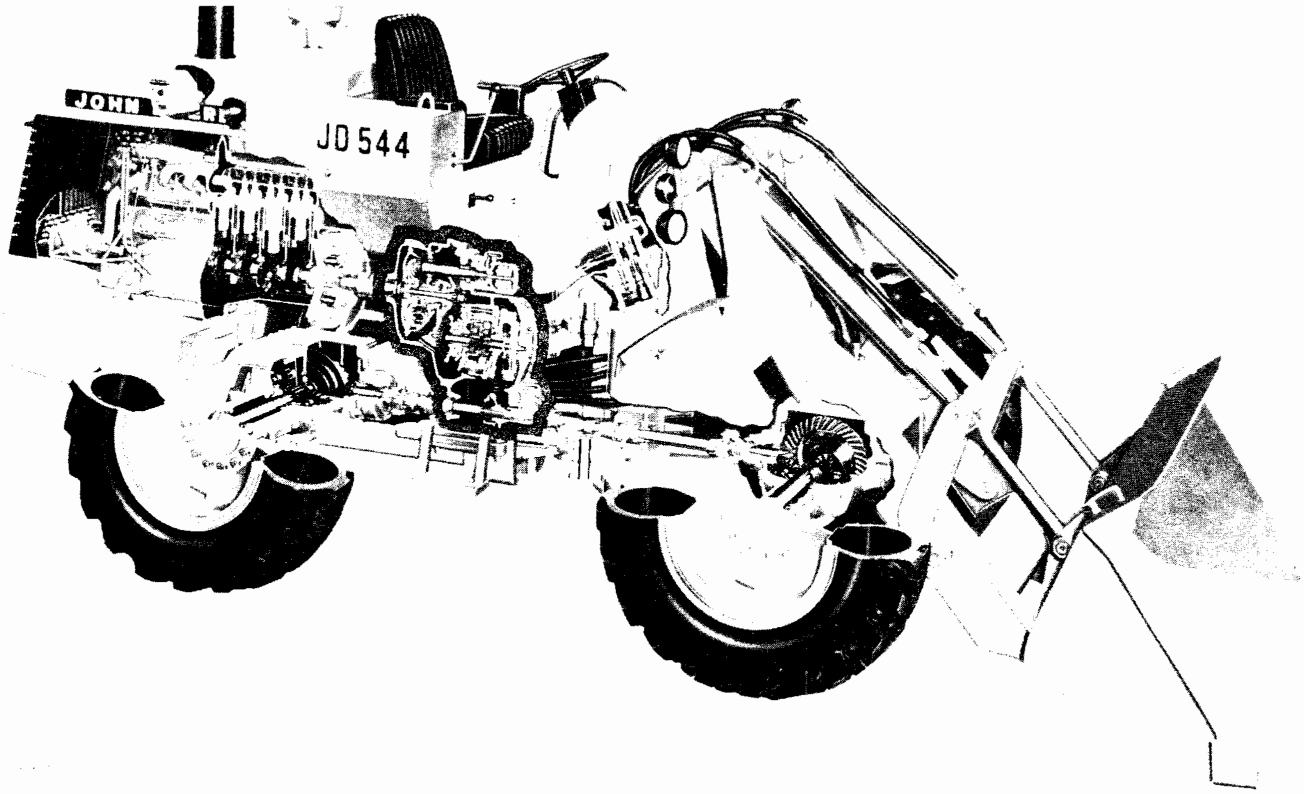


Fig. 15—Torque Converter Transmission In Four-Wheel Drive Machine

(b) Lack of cooling—plugged cooler core, low coolant level, defective water pump (water-cooled systems only).

(c) Low fluid level—clogged filter, excessive leakage past converter seals, restricted oil line, defective oil pump.

(d) Slipping clutches in planetary gear sets.

Third, failure of the converter may cause the overheating. Worn edges and pits on vanes of pump, stator, or turbines will reduce their efficiency.

2. Noise

Unlike overheating, which can be tested, noise is hard to explain to another person.

A new operator or serviceman may never hear a noise directly related to a converter failure.

Yet, an unusual noise heard by an experienced operator or serviceman may be the first sign of damage in the converter.

Noise from a converter malfunction may be a whining or growling and may be steady or intermittent.

Worn or dry bearings often produce a hissing noise that will develop into a bumping or thudding sound when they completely fail.

Other sources of noise are: Worn gears, worn or bent shafts, excessive shaft endplay, shafts misaligned with the engine, and worn freewheel clutches. All these noises mean a possible converter failure.

A mechanic's stethoscope is a valuable aid for isolating noises in the converter.

3. Leaks

Converter leaks can be one of two types:

- *Internal leaks*
- *External leaks*

INTERNAL LEAKS

By internal leaks we mean those *within* the converter.

As we learned earlier, the converter uses large amounts of oil at high velocity.

If oil is lost from the converter housing by leaks in the pump, turbine, or stator, a loss of power or erratic operation will occur.

Leaks may be caused by the wrong torque on converter bolts.

On some converters, the housing which covers the converter can be removed to determine if a leak has occurred in the converter. Check by starting the engine and operating the transmission until the oil leak shows up.

If leaks form around the converter cover, check the tightness of the cover bolts with a torque wrench. If this fails to correct the leak, disassemble the cover, check the machine surfaces of the cover and flywheel, and install a new gasket.

EXTERNAL LEAKS

By external leaks we mean those occurring outside the converter but still affecting its operation.

These include leaks at cooler lines, filter lines, and pressure or temperature gauge fittings.

Visually check all fittings and oil lines for leakage.

4. Machine Response

Normally a malfunction in the converter will affect the machine's response to load and speed changes.

A machine which lacks power and acceleration at low speed may have a turbine freewheel clutch failure.

Changes in hydraulic pressure, flow, and temperature also affect the performance of the converter, and thus affect the machine's performance.

If you picture the converter as in Fig. 7 (a flow of oil from a nozzle) it is easy to see how heavy cold oil, low pressure, or low flow will affect a response.

TESTING THE TORQUE CONVERTER

As in trouble shooting, *testing* is most effective when the engine, converter, and gear train are regarded as a unit, one part affecting the other.

Fig. 16—Pressure And Temperature Check Points On A Typical Torque Converter Transmission

Fig. 16 shows the points where pressures and temperature may be tested on a *typical* torque converter.

Before testing, several checks should be made.

BEFORE TESTING

1. Check the fluid level. Be sure it is not *above* or *below* the required level.

2. Start the engine and warm the engine and transmission to operating temperature.

3. Shift into each gear and operate for a minimum of 15 seconds, checking the selector valve detent positions against the related positions on the shift indicator (if equipped).

4. *Never allow the transmission to heat up beyond the maximum operating temperature.*

TESTING PRESSURE AND TEMPERATURE

Attach the necessary temperature and pressure gauges to the converter. Be sure that all tests are performed as required by the machine Technical Manual.

For example, tests made on the typical unit in Fig. 16 are as follows:

(1) Main Pressure (at full throttle—no load).

(2) Converter-Out Pressure (at full throttle—no load and at full throttle stall in high range).

(3) Lubrication Pressure (at full throttle—no load).

(4) Converter-Out Temperature (during normal operating conditions).

TRANSMISSION STALL TEST

IMPORTANT: Be sure to consult the machine Technical Manual before conducting a stall test. Some manufacturers do not recommend this test.

The stall test tells whether or not the engine, torque converter, and transmission are performing satisfactorily *as a unit.*

The test is made while the output shaft is braked and the engine is running at full throttle.

Refer to the engine-converter matched performance curve to find the acceptable speed at stall (see the Technical Manual).

Procedure

1. With the transmission warmed up, connect an *accurate* tachometer to the engine.

2. Block the machine securely and shift to the desired range.

3. Accelerate the engine to full throttle.

4. After reaching a stabilized engine speed, record the engine speed.

IMPORTANT: Because of rapid heating of the oil, never hold the machine stalled for more than a few seconds at one time.

Results

If the engine and transmission are failing, it will show up when the actual engine speed in the stall test is compared to the normal speed in the engine-converter chart.

Allow for variations in the test speed due to climate, altitude, engine accessory loads, and power input at converter, all of which affect the stall test results.

TEST YOURSELF

QUESTIONS

1. (Fill in the blanks.) *A ________ ________ can only transmit torque, while a ________ ________ can multiply the torque it receives.*

2. *What part in a torque converter directs the fluid back to the pump?*

3. *(Fill in the blanks.) Once torque demand is reduced in the converter, the pump and turbine "lock up" and________ flow is changed to________ flow.*

4. *True or false? "The stator will turn in either direction."*

5. *True or false? "Only one turbine can be used in a single torque converter."*

6. *Match each item below with the best match at the right.*

a. Oil flow	*1. Drives the turbines*
b. Oil velocity	*2. Engages the planetary clutches*
c. Oil pressure	*3. Lubricates and cools*

(Answers on page 9 in back of text.)

Machine Design
for Mechanical Technology

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at Queensborough Community College of the
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1984

Holt, Rinehart and Winston
New York Chicago San Francisco
Philadelphia Montreal Toronto London
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Fluid Couplings

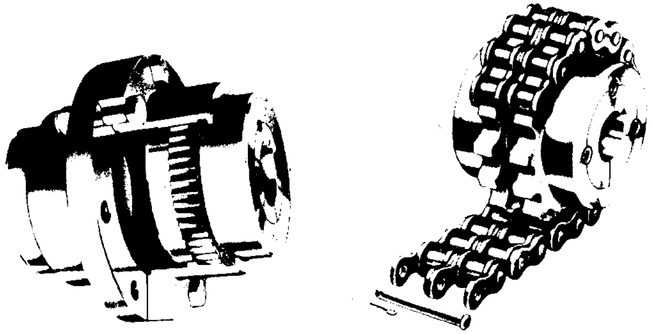


FIGURE 7.6 Gear coupling (a) and chain coupling (b). (Courtesy of Dodge Division of Reliance Electric, Mishawaka, Ind.)

Other Couplings. *Fluid couplings* essentially consist of two hollow halves, together forming a doughnut-shaped space. Radial vanes in the inside periphery serve in the input half to impart kinetic energy to the fluid, and to absorb it in the output half, except for a small portion lost to friction and shear in the fluid and converted into heat. To dissipate this heat, the housing is provided with cooling ribs (see Fig. 7.8). In order to transmit torque, there must be a certain amount of slip between the two halves (usually 3–6%).

Because fluid couplings stall at maximum output torque, they provide excellent overload protection. However, since considerable heat is then generated, extra cooling must be provided for in case of prolonged periods of stall.

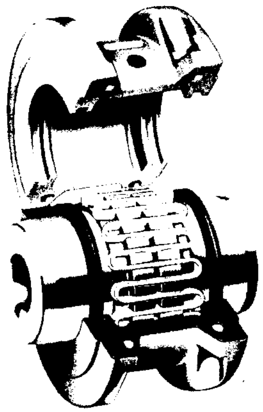


FIGURE 7.7 Falk Steelflex coupling. (Courtesy of Falk Corporation, Milwaukee, Wis.)

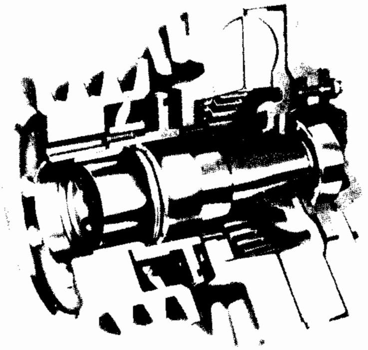


FIGURE 7.8 Sectioned fluid coupling. (Courtesy of Dana Industrial, Warren, Mich.)

The power output of the driver rotor varies with the *cube* of the rpm, so that manufacturers' rpm and hp ratings should be strictly adhered to.

By virtue of their design, fluid couplings do not permit any misalignment and only very limited float. However, their shock and vibration absorption capability is outstanding.

Centrifugal couplings or clutches engage the load when the driver has reached a certain speed. The action of centrifugal force causes driven, movable elements to press against the inside of the output half, thus providing friction for the entrainment of the driven part. The centrifugal force increases with the square of the velocity, so that at full speed adequate torque can be

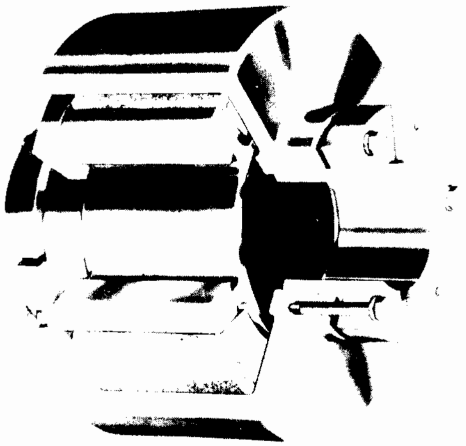


FIGURE 7.9 The Formsprag coupling. (Courtesy of Dana Industrial, Warren, Mich.)

transmitted. The operating characteristics of these couplings somewhat resemble those of fluid couplings; however, at full speed and normal loads there is zero slippage.

The *Formsprag*® *coupling* has movable elements in the shape of cylinder sectors, provided on the outside with a high-friction lining. The elements are confined between a driver spider and a follower cylinder (see Fig. 7.9).

The *Flexidyne*® *coupling* uses heat-treated shot, confined in an annular housing with a concentric disk, which is scalloped at the edge. On rotation of the housing, the shot acts as a dry fluid and settles in the outside periphery of the annulus, thus entraining and eventually holding the disk (see Fig. 7.10).

Like the fluid coupling, the Flexidyne coupling provides overload protection by slipping.

Thermal and speed-drop cutouts for protection against prolonged slipping with attendant overheating are available from the manufacturer.

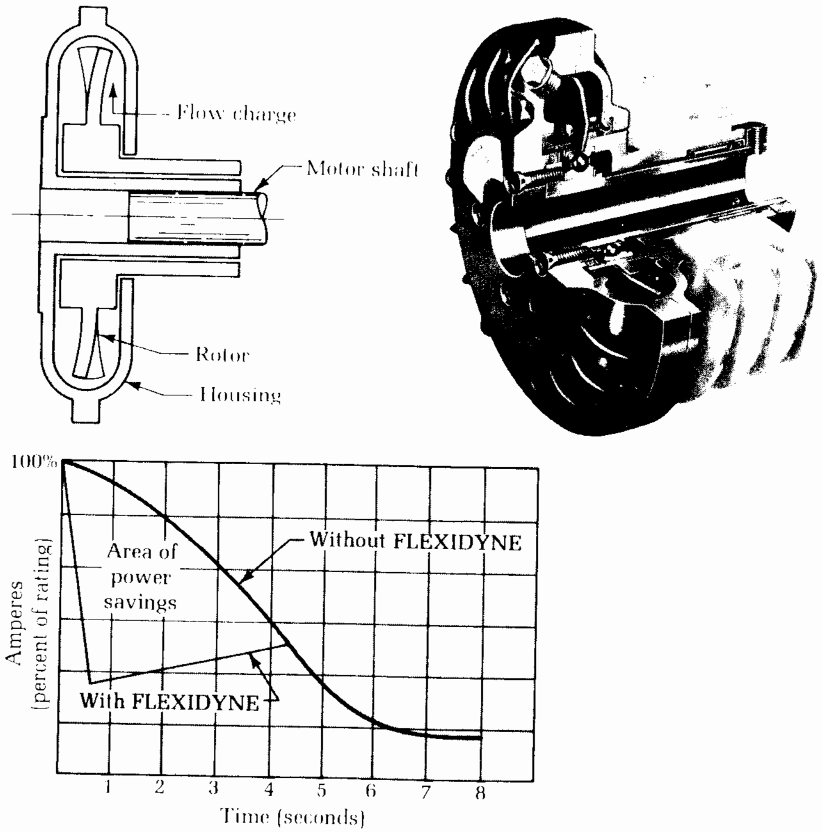
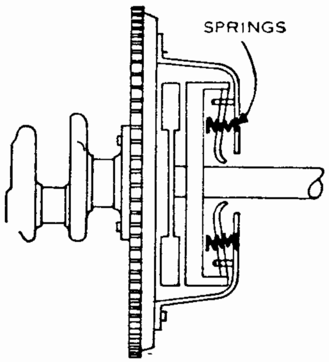


FIGURE 7.10 The Flexidyne coupling. (Courtesy of Dodge Division of Reliance Electric, Mishawaka, Ind.)

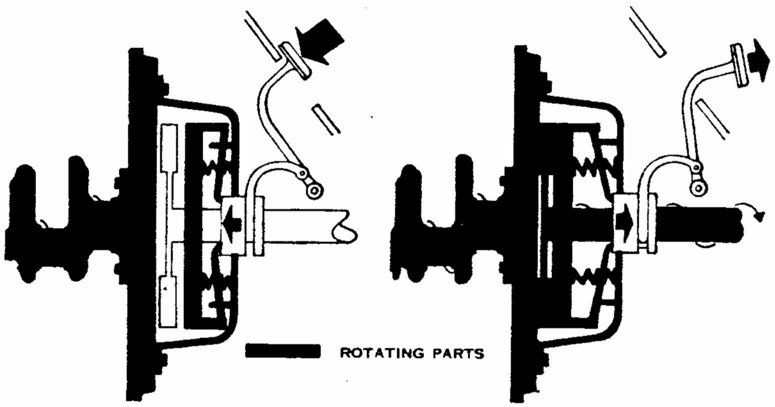


into grooves on the shaft so that they must turn together but the plate can slide forward and backward on the shaft.

A series of coil springs, or sometimes one large flat spring, act between the clutch cover and the pressure plate. They push the pressure plate toward the flywheel, squeezing the driven plate between the two. The springs are always trying to engage the clutch, and they are strong enough to keep it from slipping under any ordinary conditions. To disengage the clutch, the driver pushes on the pedal. This works through levers to pull back the pressure plate against the force of the springs. This lets the driven plate loose and disconnects the transmission shaft from the engine crankshaft.

There have been many different designs of clutches in the past, and the present ones do not all look just like what we have shown. Sometimes more than one driven plate is used, with a corresponding increase in the number of driving plates. And there are other differences. But they all work on the same principle.

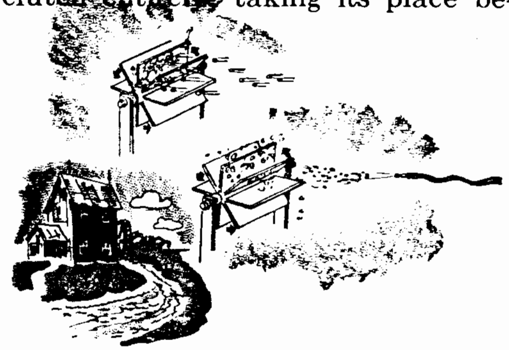
Various ways have been tried to make the clutch work



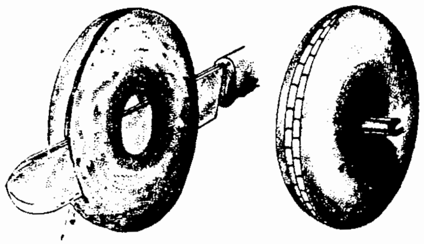
Pedal down, clutch disengaged. *Pedal up, clutch engaged.*

automatically, that is to engage and disengage without effort on the part of the driver. Sometimes vacuum power is used to operate the linkage of a standard clutch. Sometimes the clutch itself is changed to operate centrifugally. We won't go into the details of it. The principle remains the same, but centrifugal weights are arranged to engage the clutch when the engine gets up to a certain speed and disengage it when it drops below a certain speed.

There is another more common type of clutch which is really a centrifugal clutch. This is the **hydraulic coupling**, or as it is more often called, the **fluid flywheel**. Sometimes this replaces the friction clutch entirely, taking its place between the engine and transmission. In other arrangements we have both—first the fluid flywheel behind the engine, then the friction clutch, then the transmission. The fluid flywheel does not do everything the friction clutch can do, and it does some things the friction clutch cannot do. But it is a centrifugal clutch in this way—if we run the engine slowly, it will not start the rear wheels turning; when we speed up the engine, it gradually takes hold until finally the engine is driving the rear wheels with practically no slip.



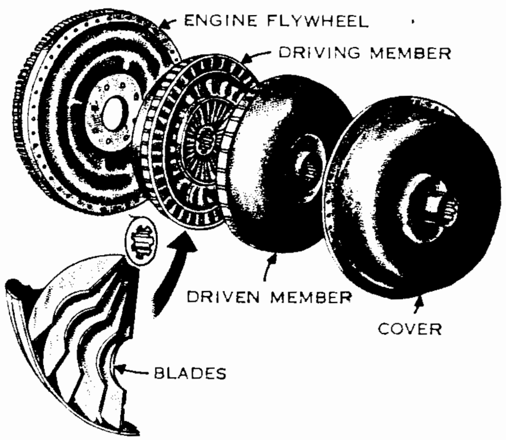
How does it work? Suppose we start with a simple example. If we shoot steel balls at the blades of this paddle wheel, each ball will give the wheel a little push, will try to turn it around. If we can shoot them fast enough and hard enough, the wheel will keep spinning. Now if we think of water or oil as being made up of a lot of small liquid balls, we can shoot these at the wheel and get the same results. You have probably seen water wheels which worked much like this, driven by the water falling over a dam or by the flow of a swift stream. That is about what we do in a fluid flywheel. But in an automobile we have

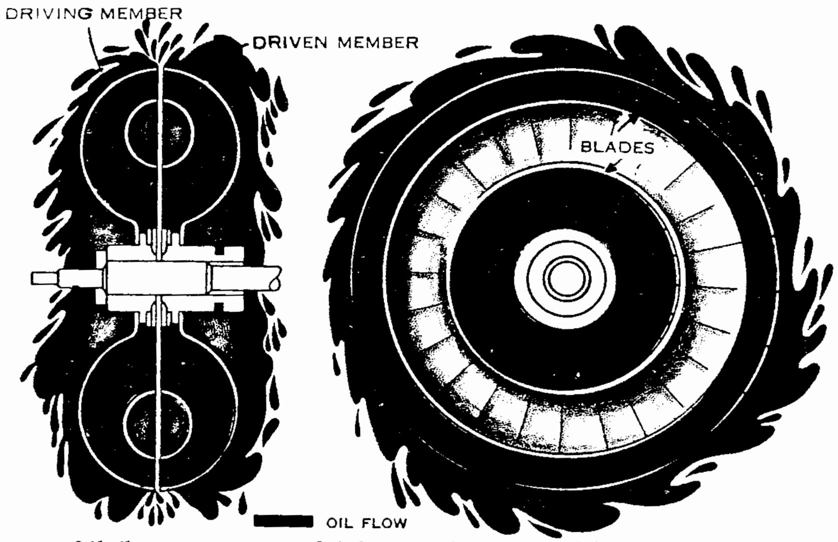


to make an artificial stream. What it amounts to is a pump forcing oil against a turbine or hydraulic motor. Many years ago it was found that the most efficient way to do this was to get the pump and motor close together, to more or less combine them. The result was a hydraulic coupling essentially the same as the fluid flywheel we use today.

The working parts of a fluid flywheel look very much like a doughnut. But the doughnut is sliced down the middle, so there is no connection between the two halves. One half is fastened to the engine crankshaft; the other to the clutch, or transmission, or some part eventually leading to the rear wheels. The doughnut is hollow, but each half has a number of straight radial blades leading from the hub to the outside edge. Very often a section of each blade is cut away, and in that space is put a metal plate or guide ring shaped like half of another, smaller doughnut. The two halves of the fluid flywheel are just alike, and when we put them together we have what looks like a skinny doughnut inside a fat one, with thin blades connecting the two.

To make this complete we put a cover around it all, the cover often being fastened solidly to one of the rotating members. Then we fill it almost full of oil. Now if the engine is running, the first half of the fluid flywheel, the **driving member,** is turning with it. If it is turning fairly fast, the oil is being thrown toward the outside of the doughnut by centrifugal force, just

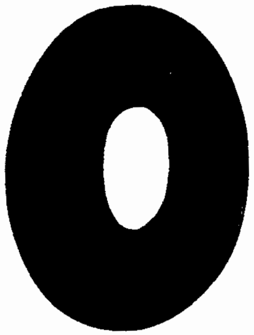




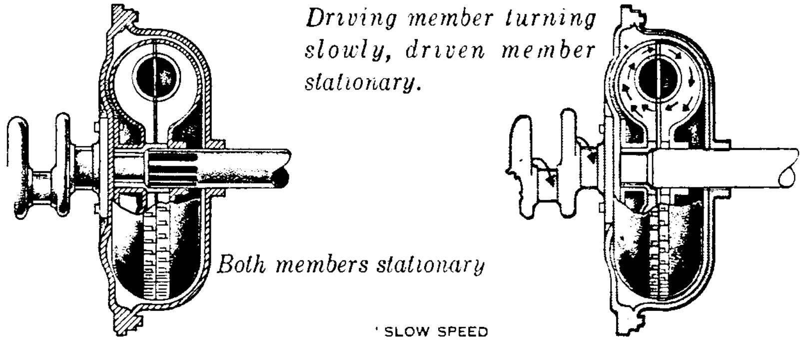
Oil flows outward in driving member, inward in driven member, and is also forced in other direction by blades of driving member.

like marbles on a phonograph turn-table. When it gets to the outside it wants to keep on going, and the only place it can go is across into the other half of the doughnut, the **driven member.**

All this time that the oil is being forced outward, it is also being whirled around in the other direction by the blades of the driving member. Consequently, when it crosses over into the driven member, it hits against those blades just as in the water wheel we mentioned and pushes them around. This tends to slow up the drops of oil, and they travel toward the hub, or center, of the driven member, then across to the driving member and repeat the whole process. Thus we have the oil continually circulating, outward in the driving member, inward in the driven member. And at the same time it is traveling in a direction at right angles to this, being pushed by the blades of the driving member and pushing on the blades of the driven member.



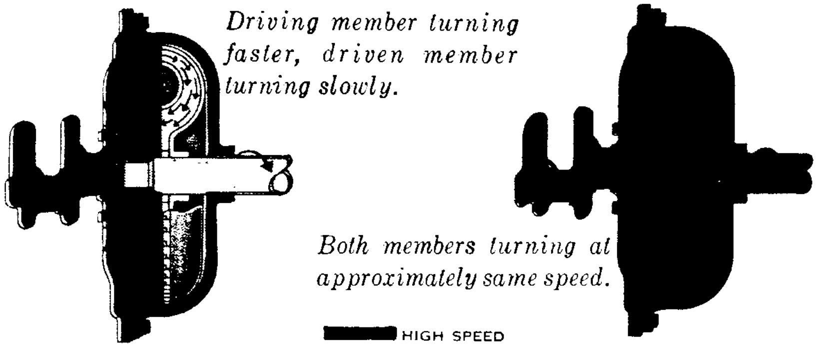
The driven member can never go quite



as fast as the driving member. There is always a certain amount of slip no matter how fast they are turning. But at ordinary driving speeds this may amount to less than one percent so it is not serious. When we get below a certain speed, however, this slip begins to get greater. Finally it gets down to the point where the driven member does not turn at all. There is still some torque being applied to it, but it is not enough to make the rear wheels turn and move the car. This means that we can stand at a traffic signal with the transmission in gear and the car will stand still just as if a friction clutch were disengaged. Then as we speed up the engine, the driven member begins to turn, gradually picks up speed and finally is running at approximately the same speed as the engine.

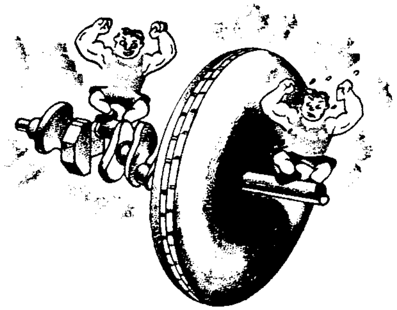
We mentioned that the driving member and driven member were just alike. There may be slight differences in them, but they are enough alike that a fluid flywheel can drive in one direction as well as the other. The oil just circulates in the opposite direction, from what was the driven member to the driving member. Thus if the car is coasting or being pushed, the wheels drive the engine just about the same as if there were a solid connection there.

The use of a fluid flywheel gives smoother pick-up and makes it impossible to stall the engine when starting or climbing a hill. It also smooths out jerks, especially at low speeds, and in some ways acts as a centrifugal clutch. As we will see later, those characteristics let us use certain types of transmissions and shift gears in certain ways



which would not be satisfactory without a fluid flywheel. But we must remember that this is just a clutch. It is not a transmission. It cannot replace the transmission because it does not increase the torque—it only transmits the torque which the engine delivers to it. We will see later on in this book some mechanisms which look very much like it and which do multiply torque. But they are different. We will point out just how they are different when we get there.

We have described the first items in back of the engine in the power train. The next major unit is the transmission, but we are going to skip that for the time being. The subject of transmissions includes several different varieties which we must consider separately to a certain extent, so we will leave them for the last. Now we will go back to the rear axle and try to show what a differential does and how it does it.



A fluid flywheel does not increase the torque

shift lever in another position it does the same thing between third and fourth. A fluid flywheel is provided, so the car can be started in third if desired and thus the driver does not need to do any shifting.

We will not try to explain the action of this transmission, but it is quite similar to that of the constant mesh type we described earlier. A positive clutch with synchronizers slides back and forth to make the shifts, but this is moved by a combination of vacuum and spring force instead of being shifted by hand.

As we said, these are only examples of automatic transmissions of various types. There are a great many others, but we cannot attempt to cover them here. We will show one more completely different type of automatic transmission, however, in the next section.

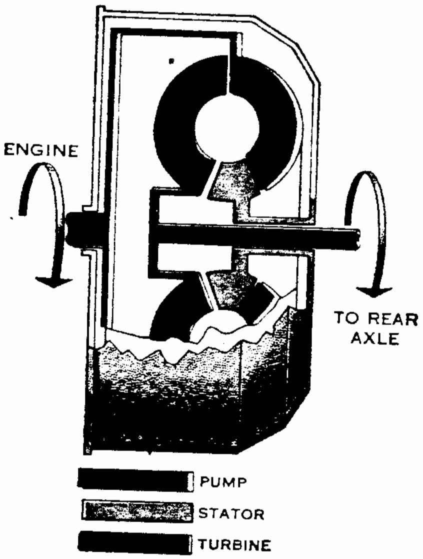
Hydraulic Torque Converter

In our discussion of the fluid flywheel, we pointed out that it was simply a clutch, a hydraulic clutch, which could not deliver any more torque than was put into it. It is a very useful addition to a transmission, but it cannot replace the transmission because it is not a torque multiplier.

But we can make a torque multiplier out of it, and the majority of automatic transmissions today use the multiplying type, commonly designated as *hydraulic torque converters*.

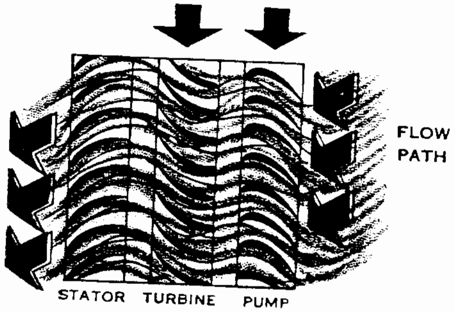
In principle, all we have to do to a fluid flywheel to make a torque converter is add another set of blades—*stationary* blades. You know the old rule that for every force there must be an equal and opposite reacting force. In transmissions this means that we cannot multiply torque unless we have some solid point to push on. We usually say we must have a reaction member, some stationary part connected to the frame of the vehicle. In a conventional transmission the whole casing is fastened solidly and this holds the shafts in place. In the planetary we have to grab hold of one of the three members before we can multiply torque—we have to hold it stationary in relation to the frame. And we have the same situation here. In a fluid flywheel the whole thing turns around together. But if we put in a new part, a set of blades tied solidly to the

frame, we have something to take the reaction, to furnish the reacting force. Then it can multiply torque.



We show here the simplest arrangement. We have the pump, or driving element, and the turbine, or driven element, just as in the fluid flywheel. But between them we add a stator, the reaction element. The casing is filled with oil, which circulates in the usual manner, outward from the pump, inward through the turbine, and then through the stator back to the pump.

The blades are not straight and flat however. If we could spread the three members out flat and look down on them, we would get an idea of their shape and how the liquid flows through them. The pump pushes the oil in the direction it is turning, and this oil hits the turbine and forces it to turn in the same direction. In doing this the oil bounces off the turbine blades in the opposite direction, and is flowing somewhat backward when it reaches the stator. If this reaction member were free to turn it would turn backward, but it is held tight. So it straightens out the oil and gets it moving in a forward direction again before it returns to the pump. In this way the motion of the oil assists the pump, and that is why such an arrangement can multiply torque.

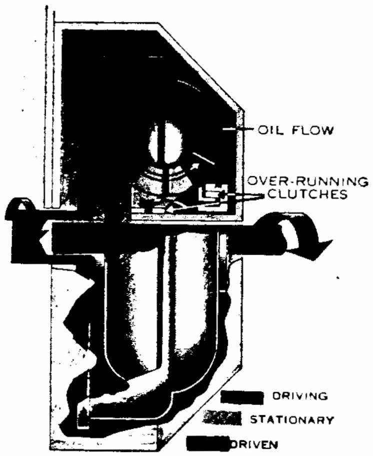


A hydraulic torque converter is completely automatic in itself. It furnishes the greatest multiplication of torque when the car is starting from standstill, and this becomes less as the car picks up speed. The torque converter does not shift. It just smoothly changes from one ratio to another and to anoth-

er in a continuous fashion, without definite steps. It is what we know as a continuously variable transmission. Where the conventional transmission changes like this , it changes like this .

This seems like the perfect way to drive a car, but there are some problems. If we tried to use the simple design we have shown, the results would be disappointing. Such an arrangement gives maximum efficiency at only one speed, and a large part of the time we would be wasting fuel. The stationary blades are necessary for multiplying torque, but when we are cruising along they just get in the way and churn up the oil. The curvature of the blades in all three elements is important, and if we design them for one condition they may not be so good for others. It is difficult to get as much torque multiplication as we desire to give good performance in starting up, and if the designer concentrates on this problem, he must sacrifice something else.

But there are various things which can be done to overcome most of these difficulties. We lose some of the simplicity, but it is well worth it. For example, we might have two pumps instead of one—or two turbines or two stators. They would probably have different blade angles and operate more or less independently, usually one taking up where the other leaves off. There are many possible arrangements along this line, depending on conditions and what results are most desired.

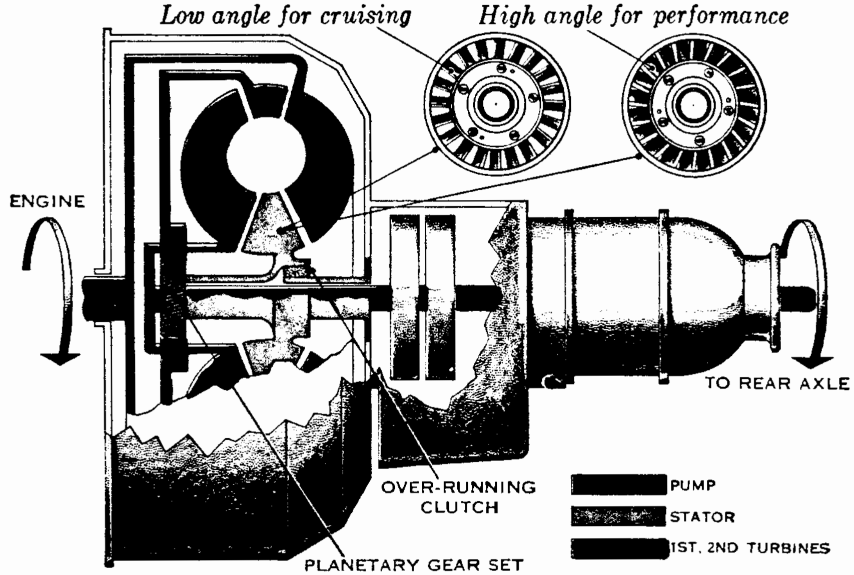


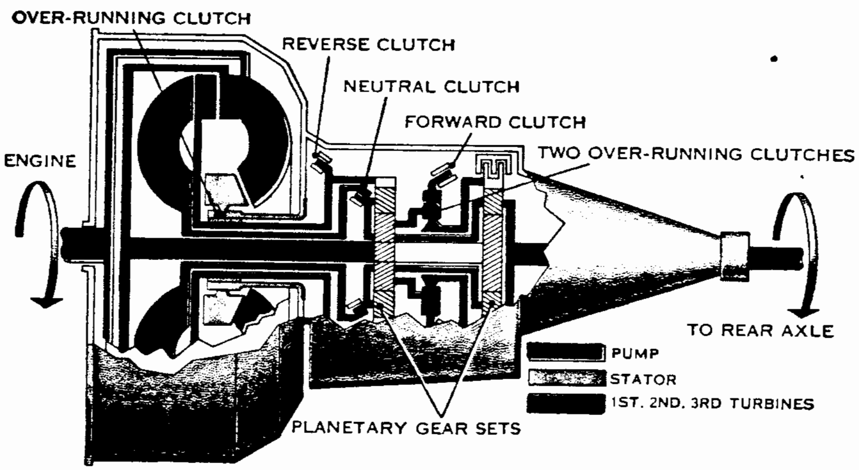
One thing which improves the efficiency is mounting the stator or stators on an over-running, or one-way clutch. This prevents the stator from turning backward, and thus it can act as the reaction element for multiplying torque. When the car gets up to speed, however, there is no backward force on the blades, and they turn forward, or free-wheel, with the oil flow. What this actually means is that the torque converter is now operating as a fluid coupling, which wastes very little power under these conditions.

In some cases the blades of the stator are mounted on pivots. Thus they can be shifted at the will of the driver from a low angle position giving economical cruising to a high angle for temporary increased acceleration or pull. This is known as variable pitch control.

With some automatic transmissions gears are used to assist the torque multiplication of the torque converter. This means that the converter does not have to be designed to give a low range ratio and is therefore more efficient over the rest of the range. It is usually a planetary gear set behind the converter. It is sometimes used only as a manually selected "low" gear. In some transmissions it is controlled automatically and used regularly for starting. Various arrangements of gears and clutches may also be used for to provide reverse, neutral and improved engine braking.

We show here an arrangement using a planetary gear in a slightly different manner. The converter has two turbines. One of these is connected to the driveshaft through a torque multiplying gear; the other is connected directly to the driveshaft. When the car is starting up, most of the force





of the oil acts on the first turbine, which drives through the planetary gear. Thus the total torque multiplication is that of the turbine multiplied by that of the gear, giving high torque for good acceleration. As the car speed increases, the speed of the first turbine also increases and more of the energy of the oil goes to the second turbine. This gradually takes over more and more of the drive, until the first turbine is contributing nothing. At this point the stator as well as the reaction gear of the first turbine begins to rotate on an over-running clutch, the planetary set is released, and the pump and second turbine operate as a hydraulic coupling. There is no "shift" at any time; it is all a gradual change just as in a simple torque converter. Thus we get the torque multiplication we need for starting and the high efficiency of the hydraulic coupling for cruising.

In another automatic transmission this has been carried further. There are three turbines and two planetary gears. The first turbine is connected to the driveshaft through a planetary gear having a torque multiplying ratio of about $2\frac{2}{3}$ to 1. The second turbine drives through the other planetary which has a ratio of about $1\frac{2}{3}$ to 1, and the third turbine is connected direct. The action is similar but stronger than in the previous case. When starting, the first

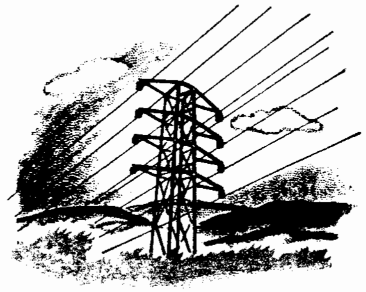
turbine is doing the driving, giving high torque multiplication through the gear. Then the second turbine gradually takes over, and finally the third one. At this point the stator and two turbines are free-wheeling and we are operating with the equivalent of a simple hydraulic coupling. In this type of transmission, the torque converter supplies complete drive ratio coverage without the need of any manual or automatic low gear.

Electric Drive

There is one other way of transmitting power in vehicles which is in common use. This is quite different from the others we have talked about, although it is for exactly the same purpose. It is the **gas-electric drive** or, what is more common now, the **Diesel-electric drive.**

Electricity is used for transmitting power in many ways and for many purposes. The network of wires spreading all over the country is simply for the purpose of getting power from one place, where it can be produced cheaply in large quantities, to a great many other places where it is going to be used. Some trains and trolley cars are driven this way, by electricity, from a central power station.

But that is not what we mean here by electric drive. And we do not mean the electric automobiles and trucks run by storage batteries, which used to be so common. What we are going to discuss are the vehicles driven by internal combustion engines in which an electrical system is used to replace the transmission and some other parts of the drive system between the engine and the wheels.



What we have is an electric **generator** and an electric **motor.** These are not little things like the generator and starter motor on our automobiles. These must handle all the power put out by the engine. (We might mention here that to the automobile or aircraft man an "engine" is an

coupler. The matter of engine design is never ending. Engineers are experimenting constantly trying to improve their power plants. The public is demanding more power, greater flexibility, economy, and smoother operation. All sorts of designs are tried out in an effort to secure the desired results. The use of fluids for power transmission is an example of developmental trends.

Engine. The automobile is a self-contained machine. It carries its own fuel and converts that fuel into power. The power developed within the engine is used to drive the wheels, and so move the car along the highways. The generator provides electrical energy which is used to charge the battery, ignite the fuel, start the engine by cranking, and light both the car and the road. The engine may be used as a brake as well as the means of developing power. Located under the hood of the motor car is a power plant, which, for completeness and efficiency, is one of the greatest mechanical products of all times. The fact that power plants are so reliable that they may be handled successfully by amateurs is their most remarkable feature.

With all the improvements that have been made in engines over many years, it is interesting to note that they still embrace the fundamental principles which first made the internal-combustion engine a success. The four-stroke-cycle principle underlies all designing. Refinement of design is the great aim of the engineers.

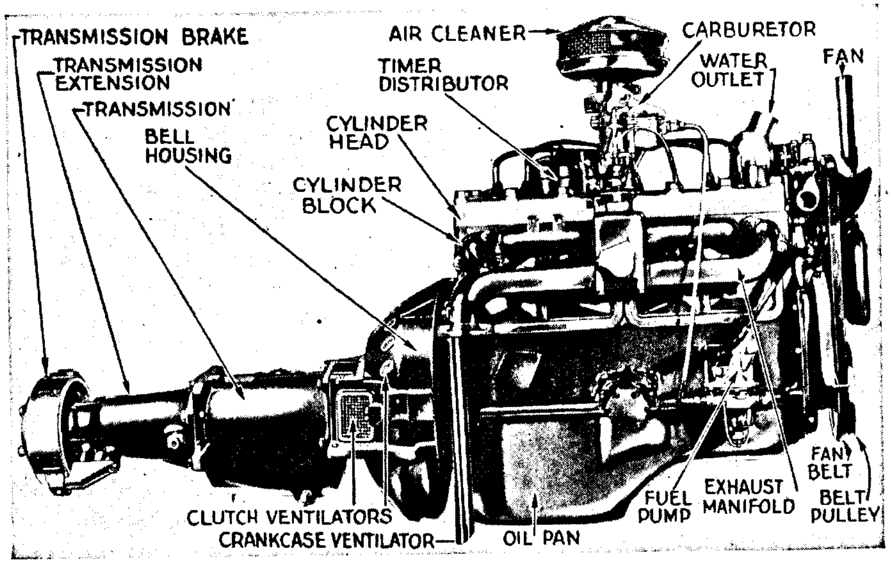


Fig. 38. Plymouth floating-power engine.

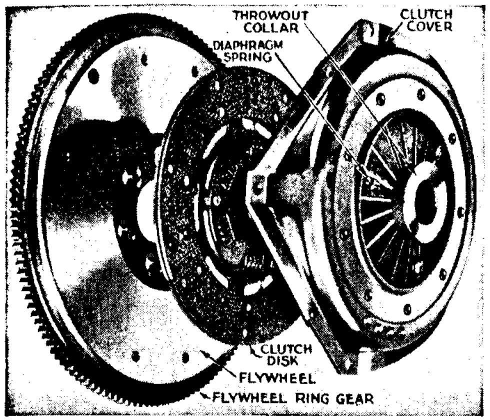


Fig. 39. Pontiac clutch.

Clutch. The power developed by the engine must be available for use at the will of the operator of the car. It must be so applied that the starting of the car is gradual. The power plant utilizing the friction clutch is so designed that, at the pressure of the foot on the clutch pedal, the power is instantly disconnected. Engaging the clutch applies the power, when the engine is running, and disengaging it, by pressing down on the foot pedal, releases the power. A plate clutch is shown in Figure 39. The clutch consists of a driving plate or plates and driven plate or plates between the driving plates. The driving plates always turn with the flywheel and the driven plates turn the transmission shaft. The fluid flywheel sometimes is used to couple the engine to the transmission. In the hydromatic automatic transmission and the torque converter transmission designs, the clutch pedal is omitted, for the reason that the fluid coupling (flywheel) takes over the functions of the friction clutch which it

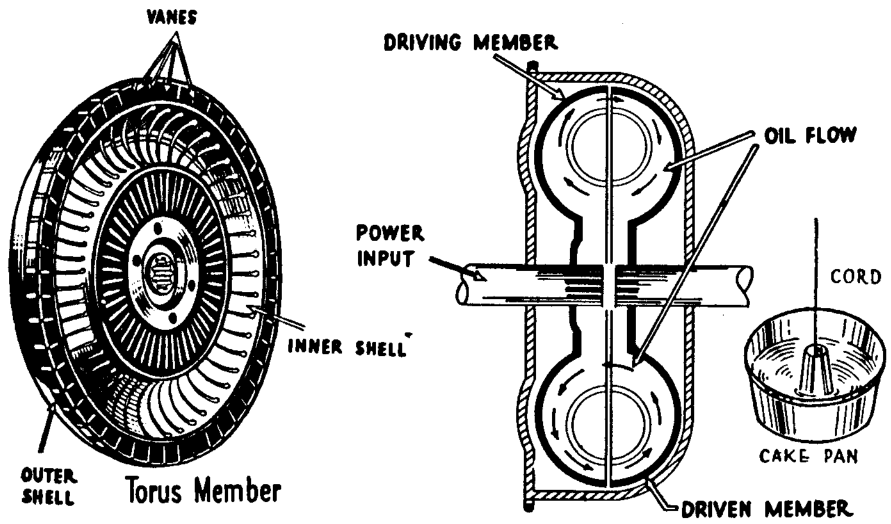


Fig. 40. Fluid coupling. Eliminates use of friction clutch.

replaces. It automatically transmits the engine power in varying degree as the engine speed increases.

Fluid Coupling. The fluid coupling consists of two units of similar construction. One of these, the driver, receives its power from the engine. Sometimes this member is part of the engine flywheel. The other is the driven member. It receives its power from the driver. These members are known as torus members. Figure 40 shows that a torus member is shaped something like a cake pan with a central hollow riser. If you were to pour water into such a cake pan and then spin the pan on its central riser, the water would be thrown to the outer edge and spill over. Now, if you were to take a similar pan and invert it over the first, the water flowing from the first would enter the second and start it spinning. Much the same thing occurs in the fluid coupling. The torus members are sealed within the flywheel in a manner to prevent the oil, which is the operating fluid, from leaking out. The driving torus member throws the oil into the driven torus member and causes it to turn. Since this driven member is on the transmission shaft, the power flows to and through the transmission. The transmission may be automatic, semiautomatic, or in the form of a torque converter. The torque-converter operation is closely tied in with the fluid coupling action.

Transmission. The transmission (Fig. 41) receives its driving power from the engine, through the clutch, and transmits or passes it on to the

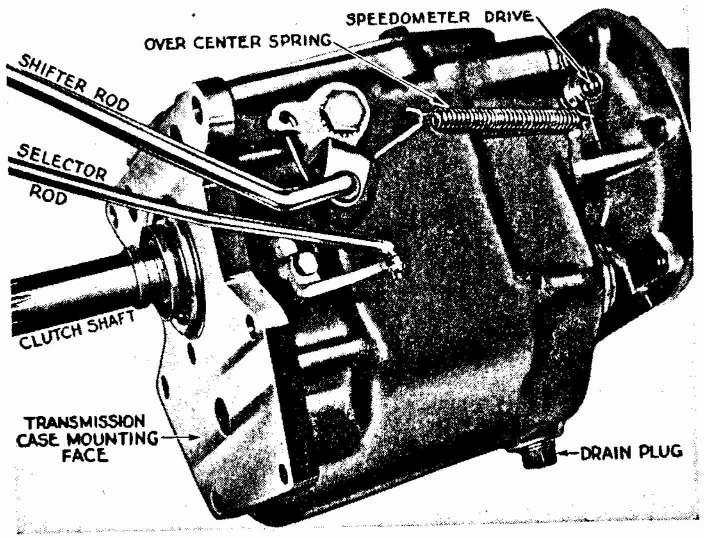


Fig. 41. Buick transmission with shift controls.

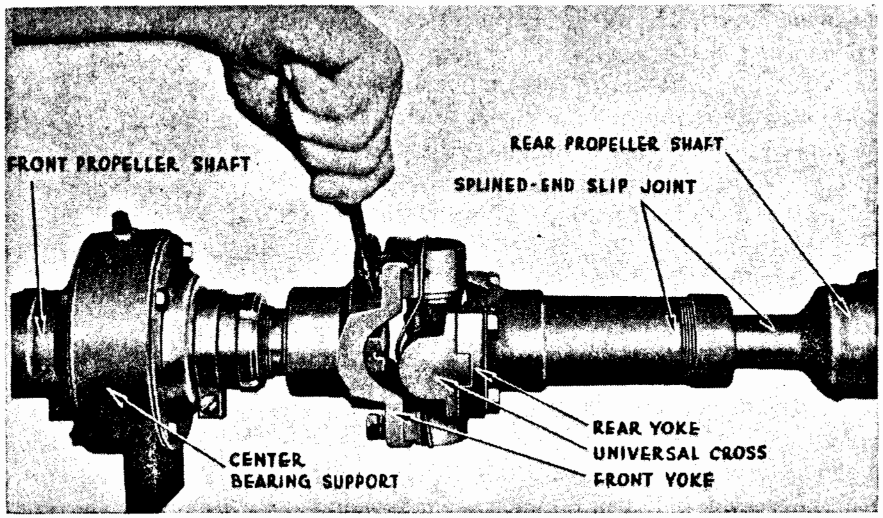


Fig. 42. Borg-Warner-Mechanics two-propeller shafts, three universals transmission to rear-axle power transmission line.

rear axle. In high speed or gear, it passes the power directly through. In other speeds, it steps down the car speed but delivers a greater torque (turning effort or power). In one gear (reverse), the direction of drive is reversed within the transmission. Transmissions have been built integral with the rear axle, that is, on the rear axle. They have been mounted amidships, that is, on the frame of the car, back of the clutch and flywheel. The practice which is currently standard for all manufacturers is to mount the transmission on the engine flywheel housing. This is called unit power-plant mounting. The transmission appearing in Figure 41 is designed for this type of mounting. Three speeds forward and one reverse is considered standard. Brakes are sometimes placed just to the rear of the case and are called transmission brakes. These usually are connected with the hand-brake lever. The speedometer drive may be obtained from the propeller shaft, back of the transmission case, or from gears within the case. In outward appearance, there is little difference as between the standard syncromesh and any of the automatic or torque-converter types.

Propeller and Universals. Figure 42 illustrates a propeller and universal arrangement. The universal receives the power from the transmission, and delivers it to the propeller shaft and it, in turn, to the rear axle. One or two universals may be used depending on the design of the rear axle. Common practice is to use two universals on exposed single-unit drive shafts and one universal on torque-tube enclosed drive shafts. In the design

CHAPTER 15

Automatic Transmissions — Hydraulic Drives

FEATURING FLUID COUPLINGS WITH GEAR DRIVES AND HYDRAULIC TORQUE-CONVERTER DRIVE

Hydraulic Drives. In Chapter 13 the use of gears for power transmission was studied. In Chapter 14 the friction clutch, by which power from the engine is delivered to the transmission, was explained. In the present chapter the use of a fluid, oil in this case, for both the above purposes is discussed.

In most discussions of fluid use for power transmission in automobiles, two points are considered. One point is the use of fluid to couple the engine to the transmission. The other is the use of the fluid to reduce the ratio of transmission speed in reference to engine speed. In the one case, the mechanical parts of the fluid coupling and the fluid itself take the place of the disk clutch and friction in coupling the engine to the transmission. In the other case, the fluid and the hydraulic transmission parts take the place of gears in bringing about a reduction of the transmission-shaft speed with relation to the engine speed, and torque multiplication.

With the hydraulic speed reducer (torque converter), the speed is reduced but the turning effort is built up. The student should fix in his mind the purposes and duties of the two hydraulic power transmitters. The fluid coupling delivers power from the engine to the transmission, while the torque converter delivers power from the engine to the transmission and at the same time steps up the turning effort (torque) delivered to the propeller shaft.

At this point it would be well to inspect the illustrations showing sectioned views of the fluid drive, Figures 508 and 510, and the torque converter shown in Figures 486 and 514-B. Before the reader makes a study of the rather intricate mechanisms involved in their operation, it would be well to try to understand the fundamental principles governing their functioning. The paragraphs immediately following deal with operating principles, and later paragraphs deal with the operation of the fluid coupling and torque-converter designs used in current automobiles.

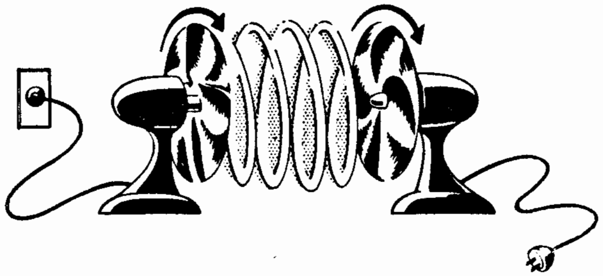


Fig. 464. One electric fan driving another is a fluid coupling. Explanatory Figures 464 to 469 by courtesy of Chevrolet.

Principle of Fluid Coupling. The fluid coupling is used in connection with synchromesh transmissions of three- and four-speeds forward design, with automatic transmissions, and with torque-converter types of transmissions. In the fluid coupling, power from the engine is passed on, not by the friction plate described in Chapter 14, but by oil thrown from a fanlike part driven directly by the engine, to a similar part which is driven by the force of the thrown oil. This principle is illustrated by the two electric fans shown in Figure 464. One fan set in front of another is driven by electric current. The second fan is not connected to the electric line. As the first fan speeds up it gradually starts the second fan turning, and in time the speed of this second fan approaches that of the driving fan. The fan producing the power might be called the driver, and the second the runner.

In the fluid coupling, the shape of the vanes in the driver (pump) and runner (turbine) are quite similar and resemble those shown in Figure 465. In electric fans, air is used to do the driving or coupling. The air is not

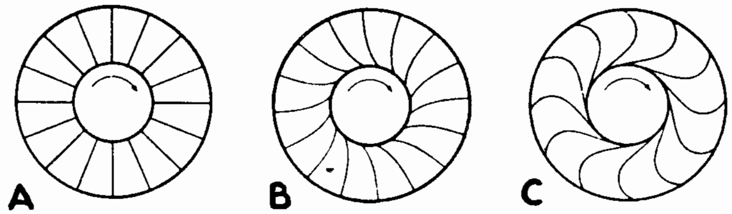


Fig. 465. Vane forms for vanes mounted in torus members. A, straight blades may be used for driving or driven fluid coupling members; B, curved blades may be used for pump (driving) members; C, curved blades may be used for turbine (driven) members in torque converter.

confined. In automotive transmissions, the fluid used is oil which is confined within the housing provided. It is used over and over. From the pump (driver) turned by power from the engine, the oil is thrown into the turbine (runner), which is thus coupled to the flywheel, in speed relation, much as is done by the friction clutch. That is, the act of coupling is not instantaneous. In a friction clutch, there is some slipping until the speed of the driven plate is brought up to that of the driving plate and flywheel. At best, this action may be somewhat uneven and may result in jerks delivered through the transmission to the rear axle. The advantage of the fluid coupling, where the driven member, the turbine (runner), is coupled to the driver only through the oil fluid medium, is that there can be no roughness or jerking. Because of their shape, the driving and driven members are sometimes called torus members. A torus is a form generated by rotating a circle in a plane about an axis. Figure 466 shows a typical driver and runner for an automotive fluid coupling.

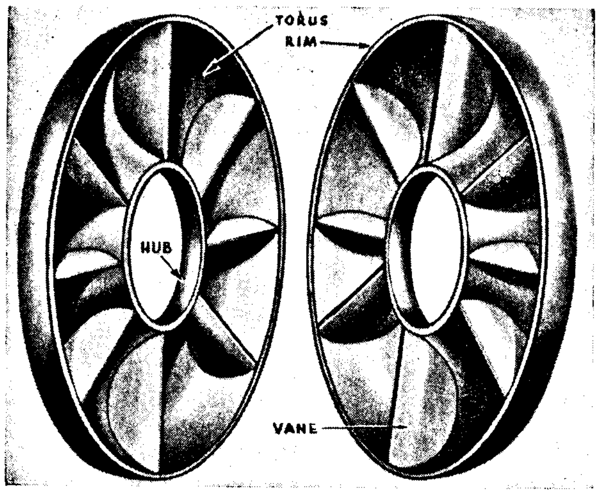


Fig. 466. Typical fluid-drive coupling driving and driven torus (donut shaped) members with straight vanes.

Different names are applied to the driven and driving parts in the various fluid drives now in use. The reader will need to associate the names with the purpose served by the parts. In some designs, the driving member is ahead of the driven member, and in others it is to the rear of the driven member. A further complication will be noted when it is understood that in the torque converter the driving and driven parts are used with reference both to torque conversion and fluid coupling.

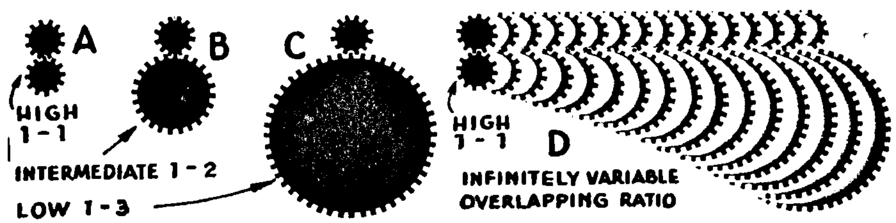


Fig. 467. A, B, C, conventional 3-speed transmission; D, torque converter overlapping gear-ratio variable from 3–1 (approximately) when starting to 1–1 (approximately) at conventional high-gear car speeds.

Torque Multiplication. Engine torque is rated in foot-pounds. Foot-pounds is the turning power of the engine measured 1 ft. from the center of the crankshaft or flywheel. This, for an engine developing about 100 horsepower at 2500 r.p.m. would be roughly 200 lb. When this torque is delivered through the rear-axle pinion to the ring gear the power available to twist the rear-axle shafts is about 800 foot-pounds. This one reduction is sufficient for high-gear operation on level roads after the car has been started. For starting, hills, and bad going, it is necessary to further step up the torque, and this is done by means of speed reductions within the transmission. These speed reductions were formerly always obtained by the introduction of gears of different sizes. In Figure 467, A, B, and C represent the conventional transmission driving gears. In high, the ratio is one to one. In intermediate, the ratio is two of the engine to one of the transmission shaft, and, in low the ratio is three to one. This 3-to-1 speed reduction with the 4-to-1 reduction of the rear axle gives an over-all speed reduction of 1 of the rear wheels to 12 of the engine. While speed is reduced, turning effort or torque is built up so the torque multiplication

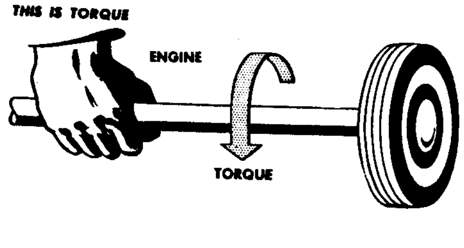


Fig. 468. Torque is the twisting force coming from the hand to the axle shaft and delivered to the wheel in a 1–1 ratio.

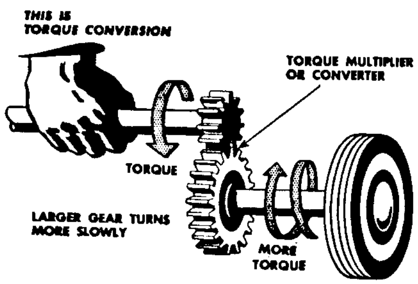


Fig. 469. Torque conversion is the use of some means (2–1 gears in this case) to step up the twisting power.

When the manual-control valve is closed, the oil from the pump is by-passed through the pressure-control valve, back to the oil sump. When open, the oil flows to the shift-valve position and to the governor. As speed increases and the desired shift point is arrived at, the governor is set to open as the governor weight moves out, and oil then flows to the shift valve. When the oil pressure matches and overcomes the spring pressure, the shift valve moves to open the oil line and allow oil to flow through the valve and through the oil line to the unit to be operated. Figure 475 shows an oil-operated planetary gear clutch.

Another such unit is pictured in Figure 477. This is a band over a drum of the internal gear of a planetary gear train. The cylindrical unit at the left is called a servo. It consists of two pistons within a cylinder having a central wall. If hydraulic pressure is admitted to the cylinder at A, the piston moves to compress the spring and moves the piston rod to push on the brake-band gear. This stops the internal-gear drum as it tightens (brakes) on its outer surface. To release the internal gear, oil pressure is introduced to both B and C. These two surfaces, under pressure equal to that of A plus the force of the spring, serve to release the band on the drum. To gain further information on equalized hydraulic pressures, the hydraulic section in Chapter 18 should be studied.

Returning to a consideration of setting the clutch plates of the planetary gear train by means of hydraulic pressure, it is evident that, if hydraulic pressure cylinders were introduced at points A in Figure 475, they could be made, through proper valving, to induce automatically sufficient pressure to force the clutch pressure ring against the clutch plates. In this way, the internal gear and the planet carrier would be locked together automatically, rather than by manual operation of the lever shown in Figure 474.

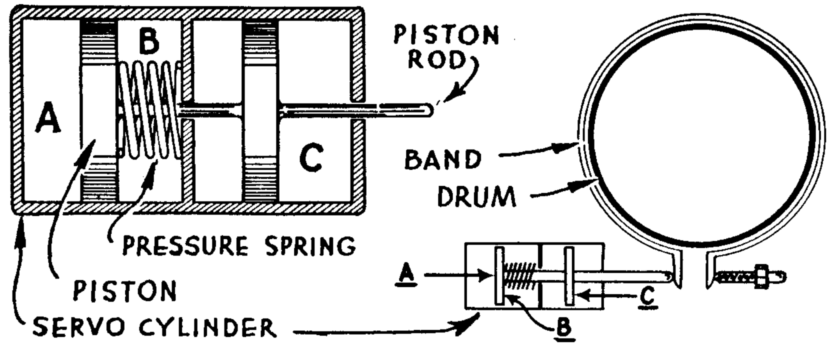


Fig. 477. Hydraulic-pressure cylinder and schematic diagram of mechanism for setting band to internal gear drum.

The foregoing discussion of the design and operation of the automatically controlled planetary gear-train mechanism gives much of the information needed for understanding the fundamental design and operation of the gearshift features of the automatic transmissions. As you study the different designs in which planetary gears are part of the structure, you can see how the design fits into the needs of the automatic operation. The control mechanism shown in Figure 473 represents but a small part of the total required for full automatic control.

Besides what has already been learned about the performance of the fluid coupling, there remains the problem of learning how the torque-converter parts function to give speed reduction and torque increase without the use of gears.

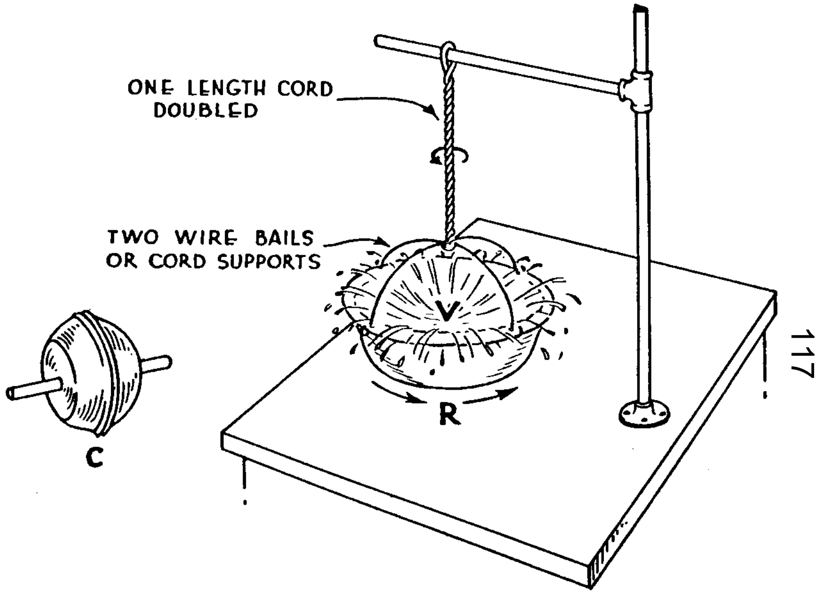


Fig. 478. Water in a whirling basin torus has two motions: R, rotary with direction of spinning basin, and V, vortex, from center outward over edge of basin. C, two basins on axles facing each other are similar to fluid-drive torus members.

Torque-Converter Speed Reductions and Power Increase. To understand fluid movement within the hydraulic-drive components of the torque converter, the reader should perform a simple experiment. Suspend a washbasin as shown in Figure 478. Fill it nearly full of water. Carefully wind the basin about its suspending cords until they are quite tight. Release

the basin, and observe the action of the water as the basin moves from a stand to a fast spin. At the start, the water moves with the basin but gradually tends to slip back. If there were vanes in the basin, this movement would largely be prevented, as it is in the fluid coupling parts shown in Figure 466. As the water in the spinning basin is affected by the centrifugal force, it starts to move outward and spill over the lip of the basin. As speed (rotary motion) increases, this outward movement is more and more marked until a hollow or vortex is formed in the water at the center of the basin. Thus, while the water is carried around and around (rotary motion) with the basin rotation, it is also given another movement (vortex movement) to the edge and out of the basin.

Now, another basin could be inverted over the first with the edge of it just clearing the edge of the lower, and the water would be thrown into the second basin. This would have two effects. First, the top basin would start spinning with the lower one, and second, the vortex flow would be directed to the center of the top basin, from which point it would be returned to the lower basin and re-used. The bottom basin might be termed the driver or the pump. The upper basin is the runner, or, when equipped with

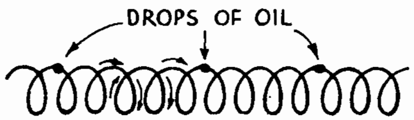


Fig. 479. Coil spring representing vortex flow.

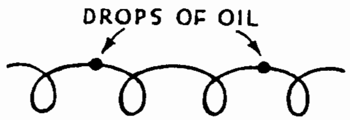


Fig. 480. Coil spring representing fluid coupling oil flow.

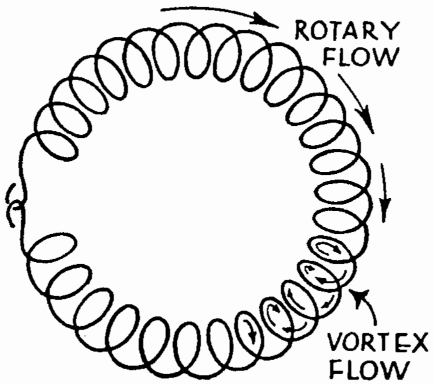


Fig. 481. Coil spring in circle representing both rotary and vortex flow of torque conversion.

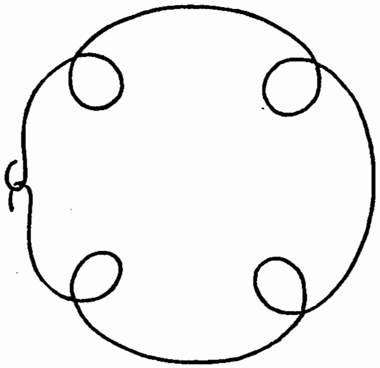


Fig. 482. Coil spring representing fluid coupling condition.

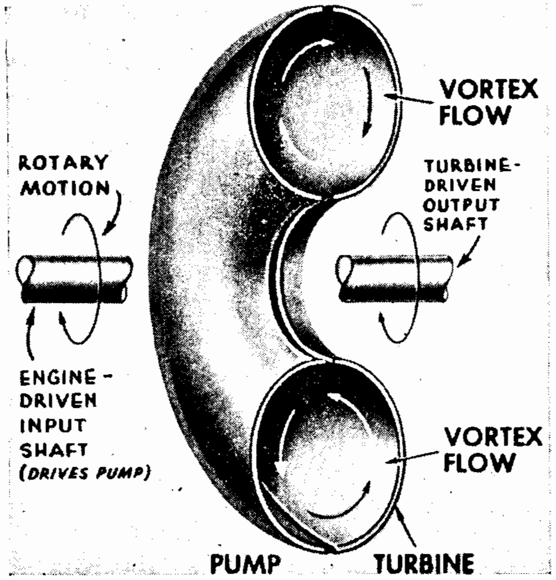


Fig. 483. Vortex flow and rotary motion.

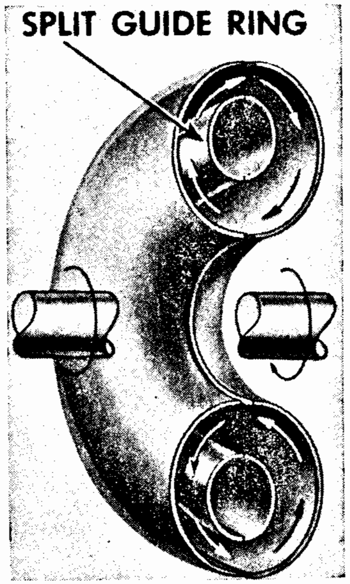


Fig. 484. Split guide ring directs vortex flow.

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vanes, the turbine. The two motions of the fluid described and observed with the basin experiment are used in all fluid drives and hydraulic torque converters. Of course, the basin or torus members are operated lip or rim in a vertical position as shown at C, Figure 478, and in Figures 466 and 483. However, since the members are housed in a fluid-tight compartment, Figure 486, and are either almost or fully filled with oil, this vertical position makes no difference in principle of action.

With the two motions, rotary and vortex, in mind, the reader can gain a fuller conception of them by studying illustrations Figures 479 to 482. Picture drops of oil on a coil spring as shown in Figure 479. The course of the drops, as they travel from end to end of the spring, is similar to vortex flow. This is especially true if the coil spring is formed as shown in Figure 481. Now, as the circular coil is rotated, the drop continues to travel about the loops, giving both rotary motion and vortex flow.

Rapid vortex flow is the power-transmitting force. High forces imparted to the oil by the engine-driven pump act against the low forces of the turbine. Marked differences in revolutions per minute of the engine-driven pump and oil-driven turbine are necessary to maintain high vortex flow. As the car picks up speed and nears the point where engine and rear-axle speeds are approximately those which would obtain in conventional clutch and gear drive, the vortex flow is more like that represented in Figures

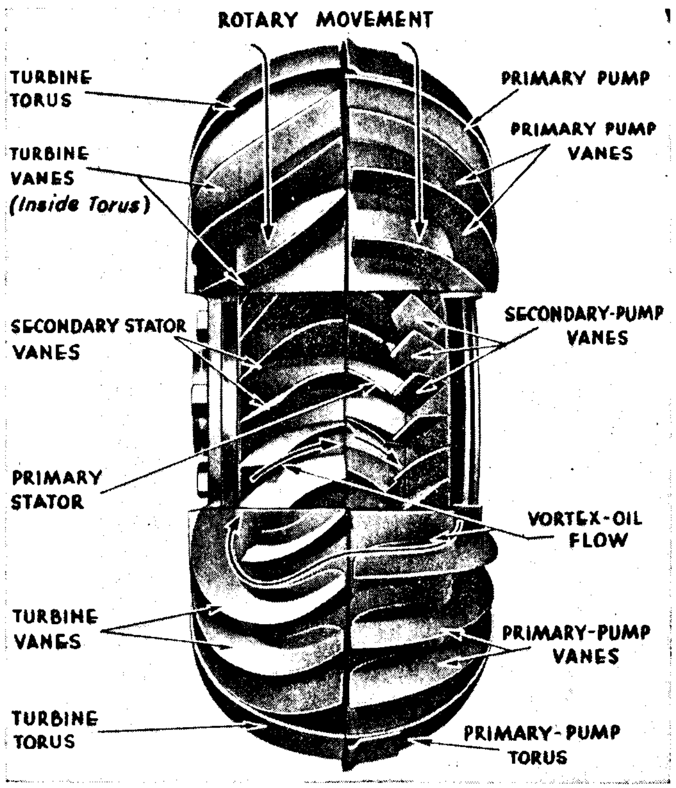

Fig. 485. Chevrolet torque converter parts with rotary movement and vortex flow.

480 and 482 which show a spring stretched out so that the coils are far apart. There is always some vortex flow in a torque-converter type of transmission, but at average driving speeds, it is relatively small compared to the very high vortex flow when the car is started from a standing position.

Figure 483 shows the two motions with reference to the torque-converter pump and turbine. Figure 484 shows the introduction of a split guide ring interposed between pump and turbine so as to give more stable direction to the vortex flow.

As mentioned previously, the pump may be located ahead of the turbine or back of it. In designing the automatic transmission, it is frequently a matter of convenience to have it to the rear as shown in the cutaway views, Figures 486 and 494, of current production models.

Figure 485 shows a cutaway view of a Chevrolet torque converter. If this illustration is studied in connection with the illustrations in Figures 486, 487, and 488, the arrangement and functioning of the parts may be better understood.

General Motors Polyphase Five-Unit Torque Converter. As adopted first by Buick and later by Chevrolet, the several designs of the General Motors fluid-operated torque converter are basically the same in principle of operation, though they differ somewhat in parts design. The essential features of the converter design consist of the five units functioning together to give the many phases of converter performance. The five basic parts of the converter are (1) the engine-driven primary pump, (2) the primary-pump-driven secondary pump, (3) the pump-driven turbine, (4) the primary stator, and (5) the secondary stator. The stators are mounted on overrunning clutches so that they may freewheel forward but may not run backward. Any attempt to do so locks them to the stationary reaction member so that in certain phases of operation they stand still within the converter and their vanes direct the oil coming from the turbine (vortex flow) back to the pump. The turbine is carried on the transmission input shaft which delivers engine power to the transmission for driving range, emergency low, and reverse speeds.

The converter unit is bolted to the flywheel and rotates within the dry flywheel housing. The converter parts form another housing within which the converter parts operate. The converter housing is filled with oil under pressure when operating. An oil seal is used on the rear of the converter to prevent oil leaking from the converter housing to the flywheel-housing cavity.

CHEVROLET AUTOMATIC TRANSMISSION

Figure 486 shows a cutaway view of the Chevrolet torque-converter type of automatic transmission. The pump operates in two phases, known as the primary and the secondary. The primary pump, to the rear of the turbine, is bolted to the flywheel at its rim. The secondary pump, Figure 487, is mounted on a freewheeling clutch on the hub of the primary pump. It may be held stationary on the hub operating at the same speed as the primary pump or, under certain conditions, it may freewheel and run at higher speeds than the primary.

The first phase of pump operation, line A, Figure 488, is used when starting the car or when under heavy loads. (Figure 488 shows an exploded view of the pump stator and turbine parts. These same parts can be identified in the sectioned view Figure 486.) In this phase, the primary pump is absorbing the full power output of the engine and the secondary pump is overrunning. This is due to the high-speed vortex oil flow to

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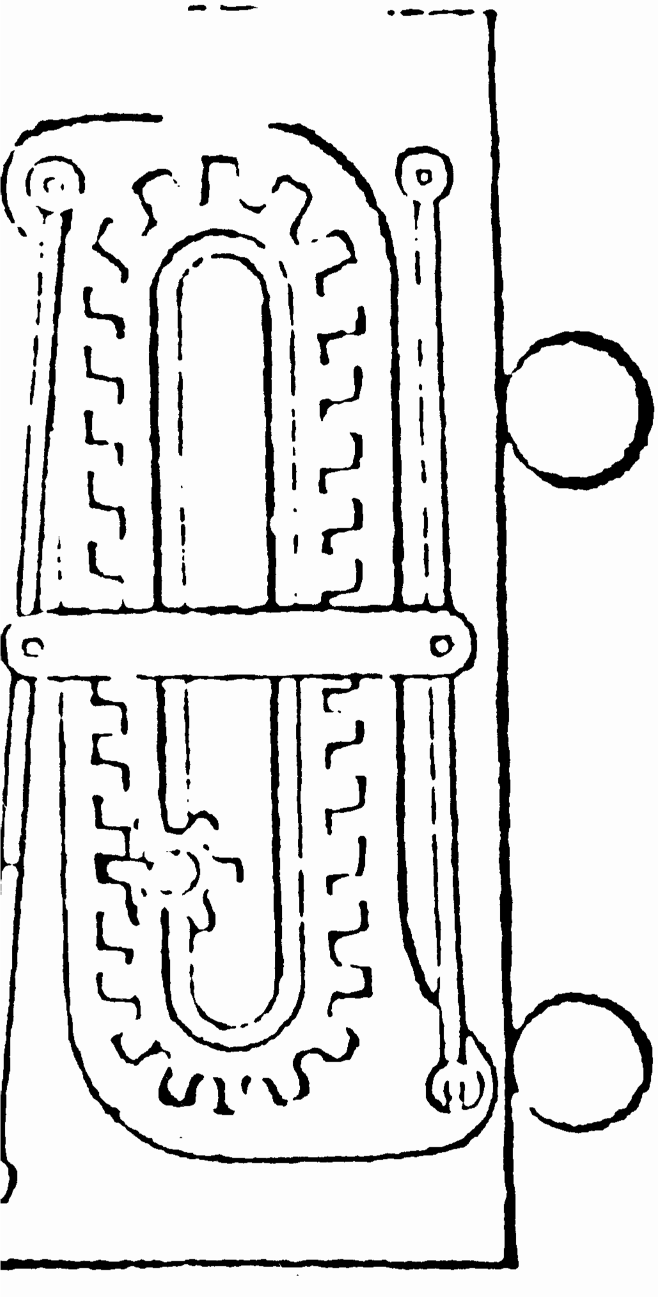
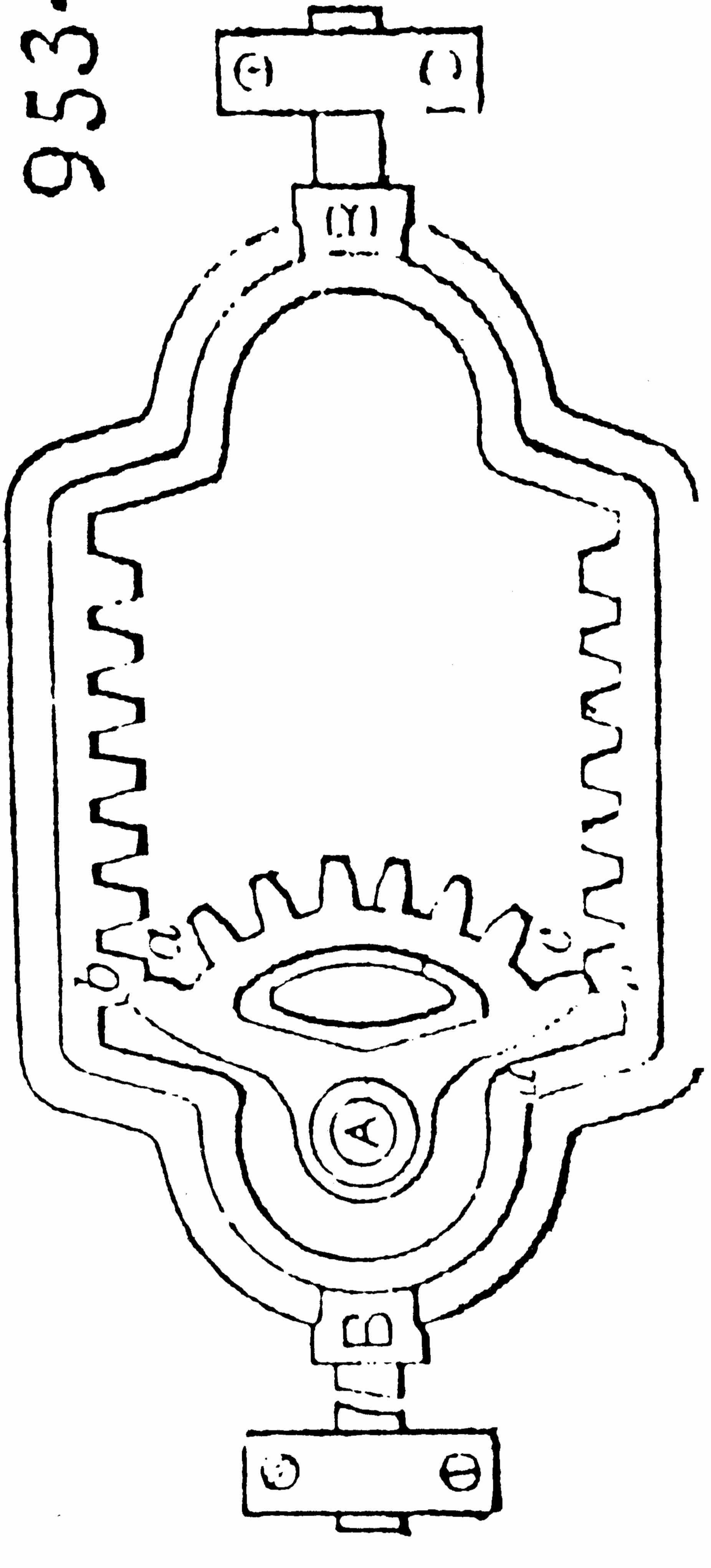




THE MANGLE RACK.—The device called the *mangle rack* is resorted to where a back and forth, or a reciprocating movement is to be imparted to an element by a continuous rotary motion.

The plain mangle racks are shown in Figs. 103 and 104, the former of which has teeth on the inside of the opposite parallel limbs, and the latter,

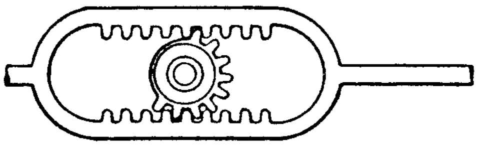


Fig. 103. Plain Mangle Rack.

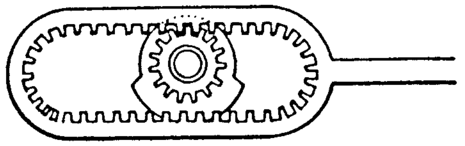


Fig. 104. Mangle Rack Motion.

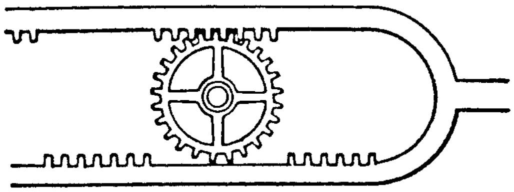


Fig. 105. Alternate Circular Motion.

Fig. 104, having teeth not only on the parallel sides, but also around the circular parts at the ends.

This form of rack may be modified so that an alternate circular motion will be produced during the movement of the rack in either direction. Fig. 105 is such an instance. A pinion within such a rack will turn first in one direction, and then in the next in the other direction, and so on.

If the rack is drawn back and forth the motion imparted to the pinion will be such as to give a continuous rocking motion to the pinion.

CONTROLLING THE PINION.—Many devices have been resorted to for the purpose of keeping the

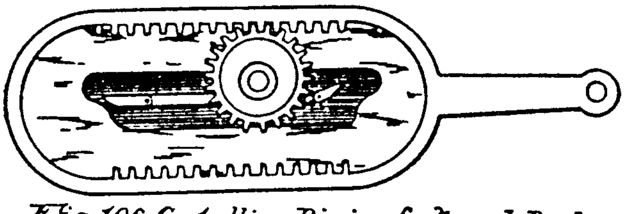


Fig. 106. Controlling Pinion for Mangle Rack.

pinion in engagement with the teeth of the mangle rack. One such method is shown in Fig. 106.

The rack A has at one side a plate B, within which is a groove C, to receive the end of the shaft D, which carries the pinion E. As the mangle rack moves to such a position that it reaches the end of the teeth F on one limb, the groove C diverts the pinion over to the other set of teeth G.

All these mangle forms are substitutes for cranks, with the advantage that the mangle gives a uniform motion to a bar, whereas the to and fro motion of the crank is not the same at all points of its travel.

Examine the diagram, Fig. 107, and note the movement of the pin A which moves along the path B. The crank C in its turning movement around the circle D, moves the pin A into the different positions 1, 2, 3, etc., which correspond with the positions on the circle D.

THE DEAD CENTERS.—There is also another ad-

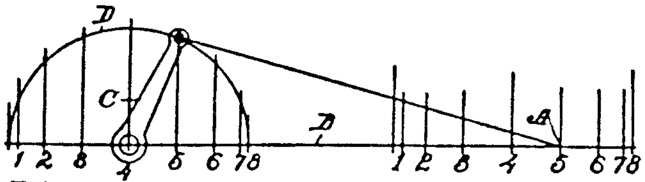


Fig. 107. Illustrating Crank-pin Movement.

vantage which the rack possesses. Where reciprocating motion is converted into circular motion, as in the case of the ordinary steam engine, there are two points in the travel of a crank where the thrust of the piston is not effective, and that is at what is called the *dead centers*.

In the diagram, Fig. 108, the ineffectiveness of the thrust is shown at those points.

Let A represent the piston pushing in the direction of the arrow B against the crank C. When in this position the thrust is the most effective, and through the arc running from D to E, and from H to G, the cylinder does fully four-fifths of the work of the engine.

While the crank is turning from G to D, or from I to J, and from K to L, no work is done which is of any value as power.

If, therefore, a mangle bar should be used instead of the crank it would add greatly to the effectiveness of the steam used in the cylinder.

Crank Motion Substitute.—In Fig. 109 the pinion A is mounted so that its shaft is in a verti-

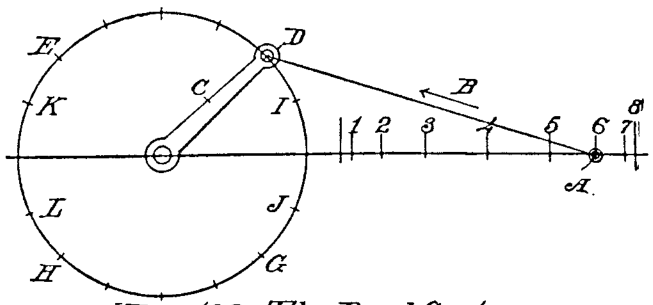


Fig. 108. The Dead Center.

cal slot B in a frame C. The mangle rack D, in this case, has teeth on its outer edge, and is made in an elongated form. The pinion shaft moves up and down the slot and thus guides the pinion around the ends of the rack.

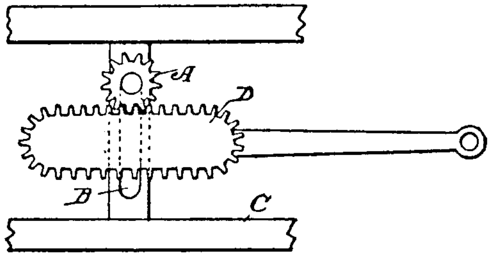


Fig. 109. Crank Motion Substitute.

The Mangle Rack.

In Fig. 102 *A* acts by rolling contact against the bar *BB* secured to the flat surface of *DE*, in which latter a groove, *FG*, is

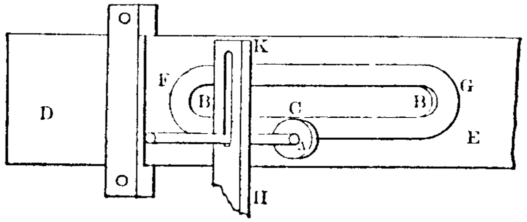


Fig. 102.

cut into which the end of the shaft of *A* is inserted, that *A* may be held in rolling contact with the barlike raised part *BB*. The ends of *BB* are rounded, equidistant from which the groove *FG* follows in its circuit about *BB*. The piece *DE* is fitted to slide in straight guides, so that as wheel *A* continues to revolve in the same direction, the bar *BB* and attached slide *DE* will be moved till *A* reaches the extremity of *B*, when further rotation of *A* will cause *A* to pass around upward at one end and downward at the other end, in continued revolving of *A* and reciprocation of *BB* and *DE*.

A bar *HK* has a slot through which the shaft of *A* passes to prevent it from swinging to the right or left.

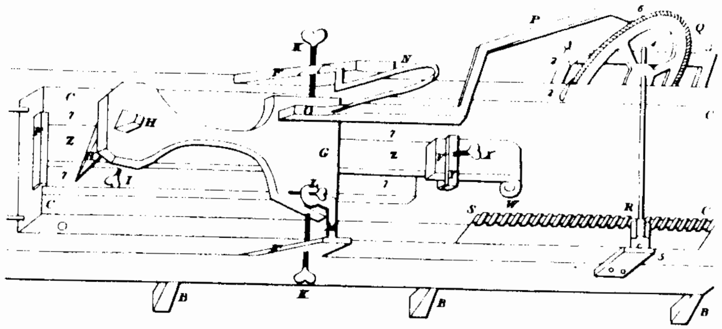
This mangle-rack movement may be given a piece *BB* of any length, without limit. The part *BB* may be narrow, even reduced to a mere line.

The velocity-ratio is the same as that in Fig. 27 or 100.

Mangle Wheels and Racks.

The pitch lines of these movements have been considered in Figs. 102 and 103, and the forms of teeth for these pitch lines present no new problems. For the treatment of the various cases of non-circular pitch lines, and teeth for the same which are likely to arise in connection with this subject, we may refer to the principles already given.

For a mangle rack with variable velocity-ratio, the pinion for the same may be non-circular when the rack pitch line is to be curved as in Fig. 39 or Fig. 46, in order that the axis of the pinion *A* may be more nearly stationary during a movement forward or back.



PECHEY'S MANGLE.

In 1822, the Society of Arts awarded to Mr Elisha Pechey a silver medal and ten guineas for an improvement in the common mangle. The novelty in this mangle is the contrivance for obtaining the alternate forward and backward motion of the box, and consequently of the rollers, by continually turning the winch in the same direction. This way of obtaining the alternate forward and backward motion may frequently be advantageously introduced in other machines; we, therefore, have extracted a description of it from that Society's Transactions.

Fig. 310 is a plan or view of the upper side of the mangle.

Fig. 311 is an end view.

Fig. 312 is a side view; the bottom parts of the frame being represented as broken off, so as to reduce it within the plate; but the whole height is seen in fig. 311.

The same letters of reference apply to the same parts in each figure.

Fig. 313 is a section of a double rack, *a a*, and *b b*, together with the plummer blocks, or guides, *c c*, (which support and guide the rack, or axis, d d, fig 313*,) being cut by a plane in the direction of, and perpendicular to, the axis.

Fig. 313* is a similar section, showing the rack in another situation, as will be hereafter described.

Fig. 314 is a view of part of the shaft, *d d*, showing the two journals, *e e*, which turn and are supported in the plummer blocks, and also the pinion, *f*, of three teeth, forming the middle part of the shaft which works into the racks.

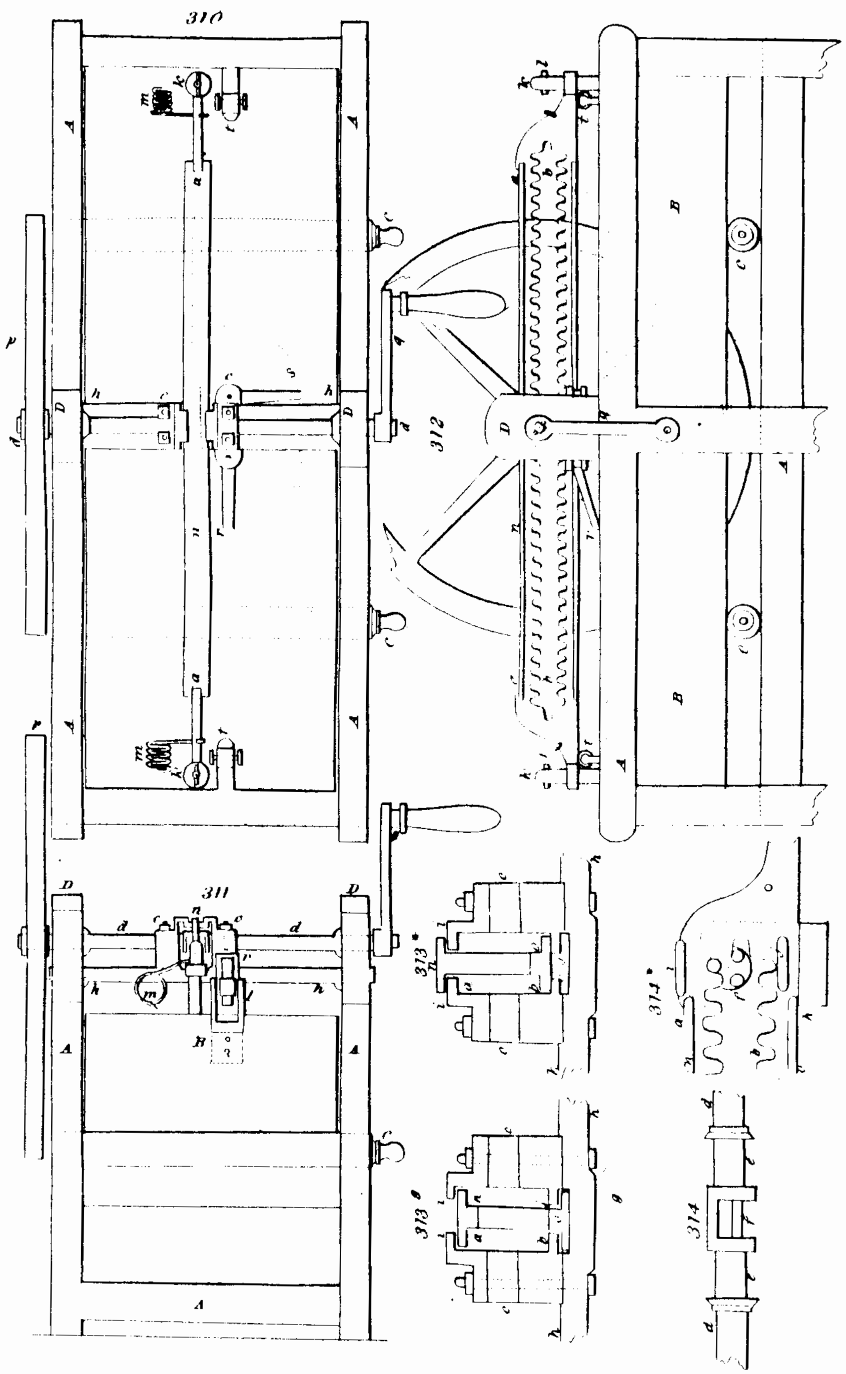
Fig. 314* is a section of the plummer block, cut in the direction of the dotted line, *g g*, fig. 313; and of the pinion, *f*, together with a side view of one end of the racks, *a a* and *b b*.

Figs. 313, 313*, 314, and 314*, are drawn double the size of the former.

A, A, &c. is the frame of the mangle, B B, is a movable box, which contains heavy weights, and C C two rollers, which are all made in a similar way to other mangles.

PECHEY'S MANGLE

Plate 41 – From 310 to 314* – Page 315



On the upper side of frame A A, &c. are fixed headstocks, D D, which are fixed and kept perpendicular to each other by an iron bar, *h h*; on the upper side of the bar are fixed the plummer blocks *c c*; the upper and lower parts of the plummer blocks have projecting pieces, *i i*, and *j j*, see figs. 313 and 313*, which support and guide the double rack; *a a* and *b b* is the double rack which is fixed parallel to the box B B, by means of two perpendicular studs *k k*; the studs pass loosely through loops in each end of the racks so that the rack is at liberty to move up and down, being at the same time prevented from sliding off by the pins *l l*, fig. 312; each end of the rack is supported by two spiral springs *m m*, fig. 310, fixed on the upper side of the box; the ends of these springs pass through small holes in the ends of the rack, and tend to support it in a middle situation, such a position as that the pinion *f*, fig. 314*, shall be in gear with the upper bar of the rack. The double rack consists of a flat bar of iron, having teeth formed in the internal part of it (as shown in figs. 312 and 314*,) and also two fins or ribs, *n* and *o*, fig. 313, fixed to the upper and lower edges of it, and projecting on each side of the rack. The use of these fins is to support the rack in two situations as shown in figs. 313 and 313*. *d d*, fig. 310, is a shaft or axis, which is supported by the head stocks D D, and by the plummer blocks *c c*, having a fly-wheel, *p*, fixed to one end of it, and a winch, *q*, to the other, by which means the machine is put in motion; in the middle part of the shaft *d d*, is the pinion *f*, shown in fig. 314, which works in the teeth of the rack; in figs. 312 and 314*, the pinion is represented working in the upper rack *a a*.

Now suppose a rotatory motion be given to the winch, the pinion will cause the rack, together with the box, &c. to move in a longitudinal direction till the end of the rack has arrived at the pinion, when it will be seen, by referring to fig. 314*, that the rack cannot pass any further in that direction. By continuing to turn the winch in the same direction as at first, it is evident that the next tooth of the pinion will take into the gap in the end of the rack, and thus cause the rack to slide up the stud *k*, till the fins on each side of the rack are raised above the projecting pieces *i i* and *j j*, fig. 313*, of the plummer blocks: the next succeeding tooth will act in the first gap of the lower rack *b b*, which will cause the rack, together with the box, &c. to move in the contrary direction till the other end of the rack has arrived at the pinion, when a tooth of the pinion will act in the gap in that end, and cause the rack to slide down the stud K, when the next tooth of the pinion will act in the first gap of the upper rack, and cause the rack to move in the direction first mentioned, and, by continuing the motion of the winch in the same direction, an alternate motion of the rack-box, &c. is effected.

To replace or change the rollers C C, one of the arms, *r*, or *s*, fig. 310, must be turned on the joint by which it is fastened to the plummer-block, in a direction parallel to the side of the rack as shown in fig. 310.

The arm forms an inclined plane, as shown at *r*, fig. 312, so that when the end of the box, B B, approaches towards the centre of the frame, the friction roller, *t*, (one of which is fixed to each end of the box), passes up the arm or inclined plane *r*, and raises the end of the box, so that the roller, C, may be removed.

Rotary Actuators

The two basic forms of pneumatic rotary actuator are:-

(i) The torque cylinder or semi-motor.

(ii) Single or paired diaphragm cylinders.

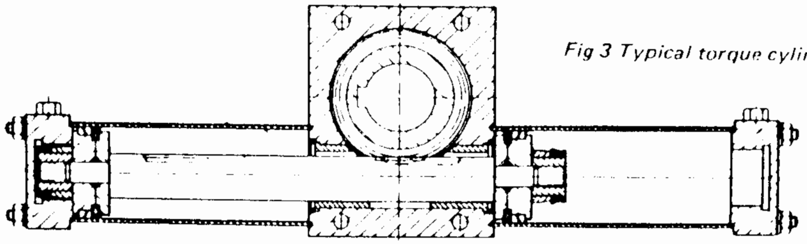


Fig 3 Typical torque cylinder in section.

The typical form of a torque cylinder is shown in Fig 3. It comprises two cylinders, back-to-back, with a common piston rod. The rod is cut with, or carries, a toothed rack engaging with a pinion mounted transversely in the centre section. Linear movement of the piston is thus translated into limited rotary output. Rotary movement is usually restricted to a maximum of 360°, or less. Nevertheless this is considerably more than that possible from a conventional cylinder connected to a lower output where the maximum angular displacement available is usually 120° or less.

A particular advantage with the torque cylinder configuration is that for constant applied pressure the torque output is constant over the whole range of rack travel. Cushioning may or may not be incorporated at each end of the cylinder, depending on requirements.

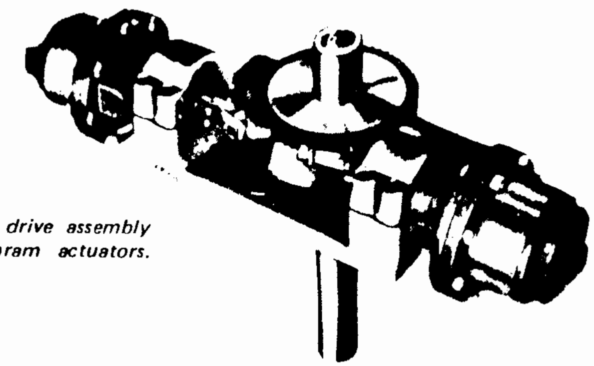


Fig 4 Telektron rotary drive assembly powered by two diaphram actuators.

A paired cylinder diaphragm-type rotary actuator is shown in Fig 4. Here two conventional diaphragm cylinders are mounted on a common centrepiece carrying a toothed wheel. Each cylinder rod terminates in a ratchet engaging this wheel. The selected diaphragm cylinder is operated in an automatic oscillating mode, driving the toothed wheel continuously in a series of progressive rotary movements. Selecting the opposite cylinder reverses the direction of motion.

Since diaphragm cylinders are essentially high-thrust devices, high output torques can be realized. Also this type of rotary actuator is capable of continuous rotation in either direction.

MECHANICAL MOVEMENTS
POWERS AND DEVICES

A TREATISE DESCRIBING

MECHANICAL MOVEMENTS AND DEVICES USED IN CONSTRUCTIVE AND OPERATIVE MACHINERY AND THE MECHANICAL ARTS, BEING PRACTICALLY A MECHANICAL DICTIONARY, COMMENCING WITH A RUDIMENTARY DESCRIPTION OF THE EARLY KNOWN MECHANICAL POWERS AND DETAILING THE VARIOUS MOTIONS, APPLIANCES AND INVENTIONS USED IN THE MECHANICAL ARTS TO THE PRESENT TIME

INCLUDING A CHAPTER ON

STRAIGHT LINE MOVEMENTS

BY

GARDNER D. HISCOX, M. E.
Author of "Gas, Gasoline and Oil Engines," "Compressed Air," etc., etc.

ILLUSTRATED BY MORE THAN EIGHTEEN HUNDRED ENGRAVINGS SPECIALLY MADE FOR THIS BOOK

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NEW YORK
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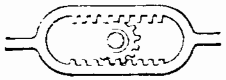
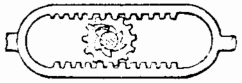
1921

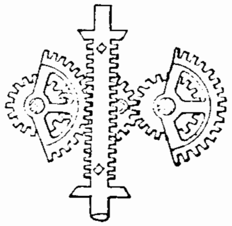


889. SECTOR PINION AND DOUBLE RACK.— Rectilinear reciprocating motion from the continual motion of a sector pinion.

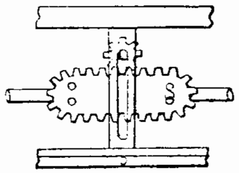
891. CRANK SUBSTITUTE, "Parson's" patent. A reciprocating double rack alternately meshing in a pinion. A cam face plate running in smooth ways in the racks and fast to the pinion lifts the racks into and out of gear alternately at the end of each stroke. The end teeth keep the pinion in mesh.



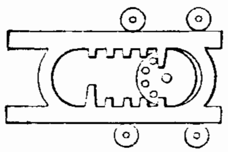
893. CRANK SUBSTITUTE. Two loose pinions with reverse ratchets attached to shaft with pawls on pinion ratchets. Each rack meshes with reverse pinion for continual motion of shaft. Many variations of this device are in use.



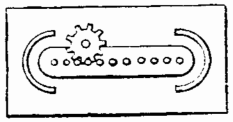
896. RECIPROCATING RECTILINEAR MOTION of a double rack; gives a continuous rotary motion to the central crank. Each stroke of the rack alternates upon one or the other of the sectors. A curved stop on the centre gear is caught on the pins in the rack to throw it into mesh with the opposite sector.



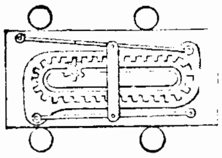
897. RECIPROCATING RECTILINEAR MOTION of a bar carrying an endless rack. A mangle device. The pinion shaft moves up and down the slot, guiding the pinion around the end of the rack.



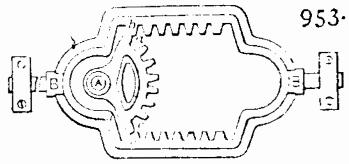
898. MANGLE RACK, guided by rollers and driven by a lantern half-pinion. The long teeth in the rack act as guides to insure a tooth mesh at the end of each motion.



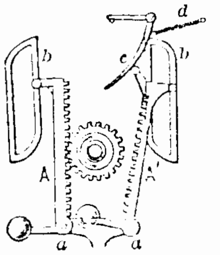
899. MANGLE RACK.—A reciprocating motion of a frame to which is attached a pin-tooth rack, the pinion being guided by the shaft riding in a vertical slot, not shown.



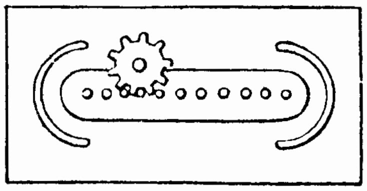
900. MANGLE RACK with stationary pinion. The rack and slot frame are jointed to the mangle box, riding in mesh with the pinion by the slot guide, leaving the mangle box free to ride and tip on the rollers.



953. ALTERNATING RECTILINEAR MOTION by the revolution of a sector by which one revolution produces both motions. The curved back of the sector just touches the extended tooth of the rack frame at *d*, while the teeth at *c* and *b* are partly in mesh with the enlarged sector end teeth, thus preventing back-lash or locking of the teeth.



996. ROTARY MOTION, from reciprocating motion of two racks alternately meshing with a gear wheel. Racks are pinned at *a*, *a*. The curved slots *b*, *b* guide the racks out and into gear. The bell-crank lever *c* and spring *d* serve to disengage the rack at the end of the up-stroke.

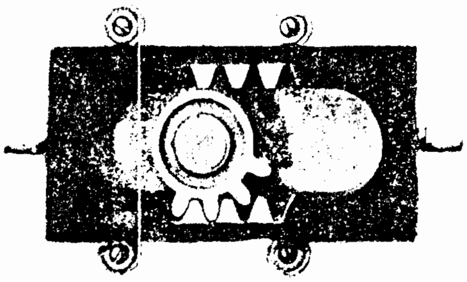


No. 204

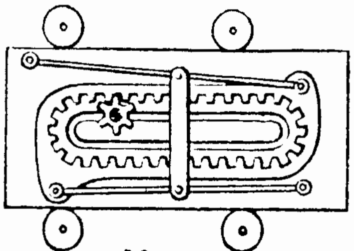
What is called a "mangle-rack." A continuous rotation of the pinion will give a reciprocating motion to the square frame. The pinion-shaft must be free to rise and fall, to pass round the guides at the ends of the rack. This motion may be modified as follows:—If the square frame be fixed, and the pinion be fixed upon a shaft made with a universal joint, the end of the shaft will describe a line, similar to that shown in the drawing, around the rack.

A Manual of Mechanical Movements
Clark 1943

79 ROTARY INTO RECIPROCATING MOTION

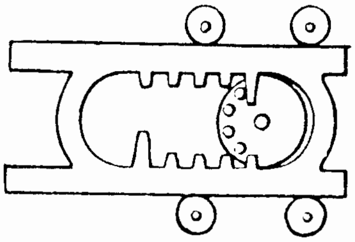


79. ROTARY INTO RECIPROCATING MOTION. The constant rotation of the four-toothed wheel inside of a specially constructed rack gives the reciprocating motion to the shaft through alternate meshing of the teeth on rack.



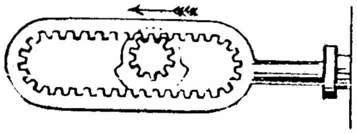
No. 205

A modification of 204. In this the pinion revolves, but does not rise and fall as in the former figure. The portion of the frame carrying the rack is jointed to the main portion of the frame by rods, so that when the pinion arrives at the end it lifts the rack by its own movement, and follows on the other side.



No. 206

Another form of mangle-rack. The lantern-pinion revolves continuously in one direction, and gives reciprocating motion to the square frame, which is guided by rollers or grooves. The pinion has teeth only in less than half of its circumference, so that while it engages one side of the rack, the toothless half is directed against the other. The large tooth at the commencement of each rack is made to insure the teeth of the pinion being properly in gear.



No. 207

C. Parsons's patent device for converting reciprocating motion into rotary, an endless rack provided with grooves on its side gearing with a pinion having two concentric flanges of different diameters. A substitute for crank in oscillating cylinder engines.

Barsanti and Matteucci.—At about this period, however (1860), and especially after the production of the Lenoir and Hugon engines, three defects had come to be recognised as the inevitable results of an explosion at each to and fro stroke of the piston. The heat generated was so great that it had to be carried off as quickly as possible, and, even with water jackets to the cylinder, parts of the engine sometimes became red hot. It was also impossible, in a double-acting engine, to compress the gas and air before ignition; and lastly, expansion of the gases was greatly limited. The stroke of the piston was too short to utilise to the full the expansive force produced by the explosion, and the products of combustion were discharged at a pressure much above atmospheric. In this way almost all the heat generated by the ignition and explosion of the gases was wasted. Many experiments were made, and many engines constructed, before it was realised that the greater the amount of heat utilised by doing work on the piston, the lower would be the temperature and pressure of the gases at discharge, and the less heat would be wasted. The next engine, invented by two Italians, Barsanti and Matteucci, showed a better knowledge of the principles of economy. In it a distinct step in advance was made, and an important principle exhibited for the first time in practice—namely, the use of a free piston, and unchecked expansion of the charge. For this reason their motor deserves attention and study, though, like many others, it was not a practical working success.

Two patents were taken out by Barsanti and Matteucci in England, the first in 1854, the second in 1857. In the first the free piston was supplemented by a lower auxiliary piston immediately below it in the same cylinder. An outline drawing of the engine is shown at Fig. 2. A is the cylinder and P the motor piston; p is the auxiliary piston, S the flat slide valve actuated by a lever F connected with the rod E of the auxiliary piston, which passes through the bottom of the cylinder. The crosshead at E is attached by two levers, not shown in the drawing, to the wheel D and the crank J, driven from the main shaft, but not revolving so rapidly. As soon as the free piston P has reached its lowest position, p begins to descend, and air is admitted between the two pistons through the passages a, b, c of the slide valve S. As the auxiliary piston descends, the slide valve is lowered with it by the lever F, the air port a is closed, and the gas port d uncovered, admitting gas to the cylinder between the pistons through d, b, and c. The slide valve next shuts off d, when the mixture is fired by a series of electric sparks, the circuit being put on by the lever F. The piston P, which has been at a stand, is now projected upwards, free of the crank shaft, and p is forced still lower, driving out the products of combustion below it through the openings i i in the bottom of the cylinder. The pressure in the cylinder beneath the free piston is now below atmosphere, the valves i i close automatically, the channel f is uncovered, and as the piston rises communication is established between the contents of the cylinder above and below the piston p through f, e, and b. The working piston descends in the vacuum, driving out the exhaust, and the same process is repeated. This is the first instance of the slide valve, afterwards much used.

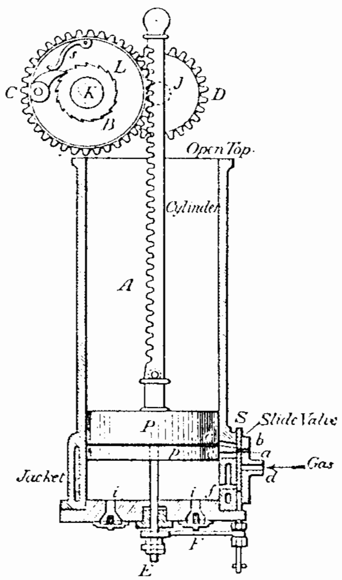


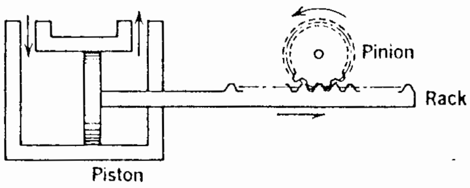
Fig. 2.—Barsanti and Matteucci's Gas Atmospheric Engine. 1854.

The arrangement of the catch is novel and ingenious. The rod of the free piston P carries a rack, and as soon as the piston begins to descend, the rack gears into the toothed wheel L, running loose on the main shaft K. The wheel L has a pawl C. As the rack falls, and drags L round to the right, the spring s presses the pawl C into the teeth of the ratchet wheel B, which is keyed on to the main shaft K, and causes B and therefore K to rotate to the right. When the piston rises the main shaft continues to turn to the right, but the movement of the wheel L is reversed; it revolves to the left with the up stroke of the piston, and C., slipping past B, loses connection with the main shaft.

In the second engine patented by Barsanti and Matteucci the auxiliary piston was abolished, the slide valve was worked by a valve-rod, and the details were much simplified. There was an auxiliary as well as a motor shaft, both having pawls acting upon the rack. In both engines a much better and freer expansion was afforded to the combustible gases than had hitherto been obtained. In fact there was no check to their expansion, except the weight of the piston, &c. But, notwithstanding its excellent cycle, this motor was never in the market, probably because the working details and the mechanism were defective. That the main lines on which it was constructed were good, is proved by the fact that they were adopted and successfully put in practice by Otto and Langen, though the German engineers appear to have designed their motor independently. The fundamental principle of the Barsanti and Matteucci engine, to utilise the whole force of the explosion in as complete expansion as possible, was excellent, and has not been improved upon. Few modern inventors have been able to approach as closely the conditions of a perfect theoretical cycle.

Analysis and Design of Pneumatic Systems   Blaine W. Anderson

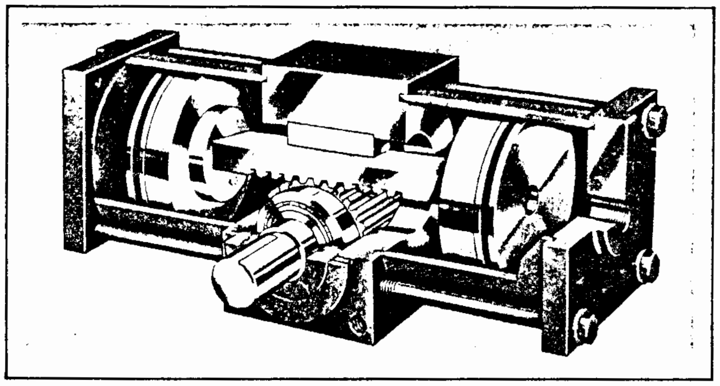
(b) Rack and pinion



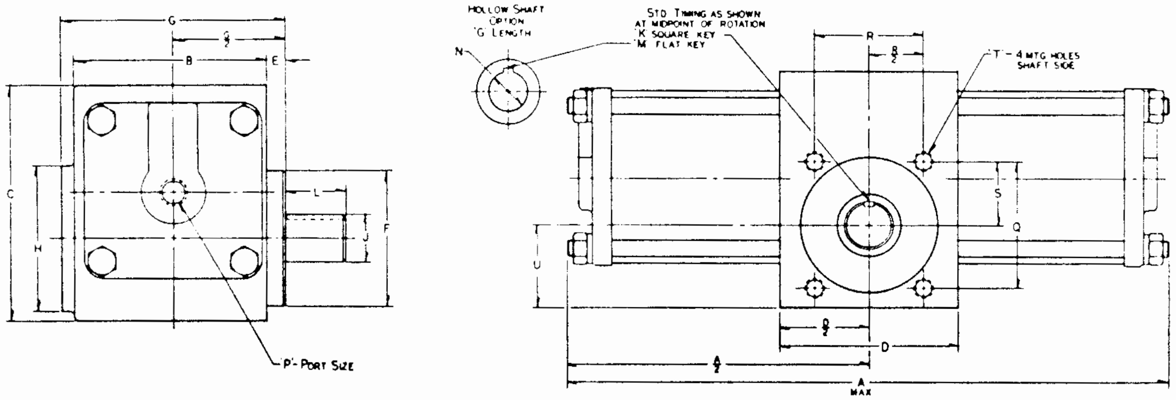
Several methods of converting linear to rotary motion are shown schematically in Figure 1.7. These are the crank arm, rack and pinion, and the helical spline or ball screw.

Series A Rotary Actuators

- Heavy Duty Air—Low Pressure Hydraulic
- Input Pressures: 25-250 P.S.I.
- Output Torques: 50-8400 Lbs. in.
- Standard Rotations: 100°, 190°, 280°, 370°
- Aluminum housing affords maximum strength and minimum weight.
- Lubricated Rack and Pinion in sealed non-pressurized chamber.



Model A 19.4-190-MS1-C at Mid-Rotation



MODEL NO.	TORQUE @ 250 P.S.I. 17.24 BAR LBS. IN.	NM	ROTATION DEGREES	DISPLACEMENT IN³	DISPLACEMENT CM³	A IN	A MM	B* IN/MM	C* IN/MM	D* IN/MM	E IN/MM	F IN/MM	G IN/MM	H* IN/MM	J IN/MM	K IN (W & L)	L IN/MM	M IN (W & H)	N IN/MM	P IN	Q IN/MM	R IN/MM	S IN/MM	T IN	U* IN/MM
A6.2	500	56.50	100 190 280 370	4.11 7.81 11.51 15.22	67.36 128.01 188.65 249.46	11.69 12.87 15.92 17.10	296.93 326.90 404.37 434.34	4⅛"	5¼"	4"	.375	2.995 3.000	4.75	3¼"	.998 1.000	¼" × 1"	1.25	3/16" × ⅛"	.875 .878	¼" N.P.T.	3.38	2.25	2.000	⅜" NC	1⅞"
A6.3	1100	124.30	100 190 280 370	9.25 17.58 25.91 34.24	151.61 288.14 424.66 561.19	11.82 13.00 16.05 17.22	300.23 330.20 407.67 437.39	104.78	133.35	101.60	9.53	76.03 76.20	120.65	82.55	25.35 25.40		31.75		22.23 22.30		85.85	57.15	50.80	× ½" UP	47.63
A19.3	1500	169.50	100 190 280 370	12.34 23.44 34.54 45.65	202.25 384.18 566.11 748.20	13.13 14.70 19.52 21.56	333.50 373.38 495.81 547.62	4⅞"	6"	4½"	.460	3.495 3.500	5.67	3¾"	1.248 1.250	¼" × 1¼"	1.53	¼" × 3/16"	1.000 1.004	OPTIONAL SAE 9/16"-18	3.25	2.75	1.625	½" NC	2⅛"
[illegible]	[illegible]	299.45	100 190 280 370	21.93 41.67 61.41 81.15	359.43 682.97 1006.51 1330.05	13.13 14.93 19.75 21.70	339.34 379.22 501.65 551.18	123.83	152.40	114.30	11.68	88.77 88.90	144.02	95.25	31.70 31.75		38.86		25.40 25.50	⅜" N.P.T.	82.55	69.85	41.28	× ½" OP	53.98
A67.4	3700	418.10	100 190 280 370	30.71 58.34 85.98 113.61	503.34 956.19 1409.21 1862.07	17.75 19.95 26.82 29.07	450.85 506.73 681.23 738.38	8⅛"	8¼"	6½"	.625	4.995 5.000	9.25	5¼"	1.998 2.000	½" × 2"	2.75	⅜" × ¼"	1.751 1.753	OPTIONAL SAE ¾"-16	4.50	4.00	2.250	⅝" NC	2⅞"
A67.6	8400	949.20	100 190 280 370	69.09 131.27 193.44 255.62	1132.39 2151.52 3170.48 4189.61	17.87 20.07 27.05 29.30	453.90 509.78 687.07 744.22	206.38	209.55	165.10	15.88	126.87 127.00	234.95	133.35	50.75 50.80		69.85		44.48 44.53	½" N.P.T. OPTIONAL S.A.E. ⅞"-14	114.30	101.60	57.15	× ⅝" OP	73.03

*Dimensions B, C, D, H and U are as cast.

The following design features and options are built into rotary actuators of both series:

Features:

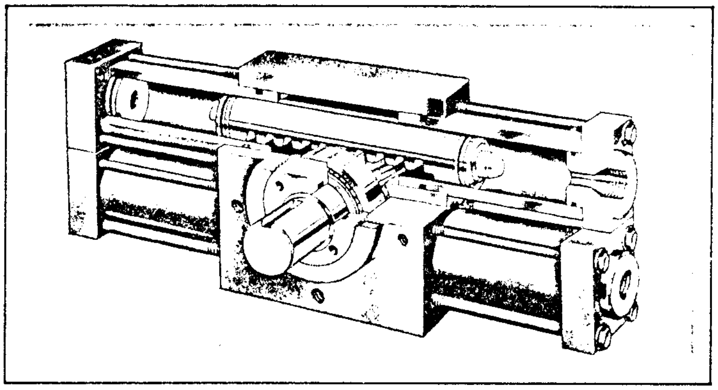
- Zero internal leakage- No drift
- Tapered roller bearings
- Pre-loaded lipseal construction piston seals
- High strength steel rack and pinion
- Heavy-wall steel cylinder tubing
- Pre-stressed steel tie rods
- Through shaft that permits customers to drill and tap for control devices

Options:

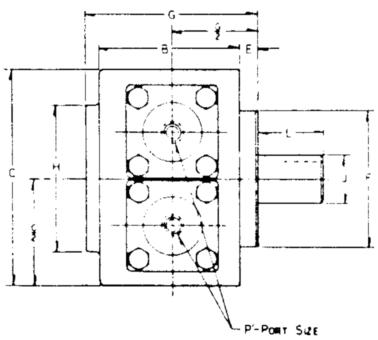
- Rotation Adjustors 0° to 20° in each direction
- Cushions
- Hollow shaft
- Double-End Keyed Shaft
- Single-End and Double-End Spline Shaft (SAE 10B)
- Multiple mounting surfaces
- Face and Bottom Flange mounting plates
- Special Ports
- Special seals and compounds
- Air Bleeds

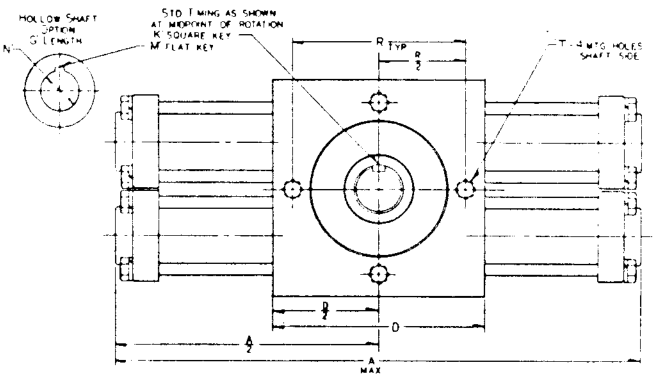
Series H Rotary Actuators

- High Pressure-Hydraulic
- Input Pressures: 100-2000 P.S.I.
- Output Torques: 5-478,000 Lbs. in.
- Standard Rotations: 100°, 190°, 280°, 370°
- Ductile Iron housing affords maximum strength and rigidity
- Oil immersed rack and pinion in sealed non-pressurized chamber



Model HC37-190-MS1-C at Mid-Rotation

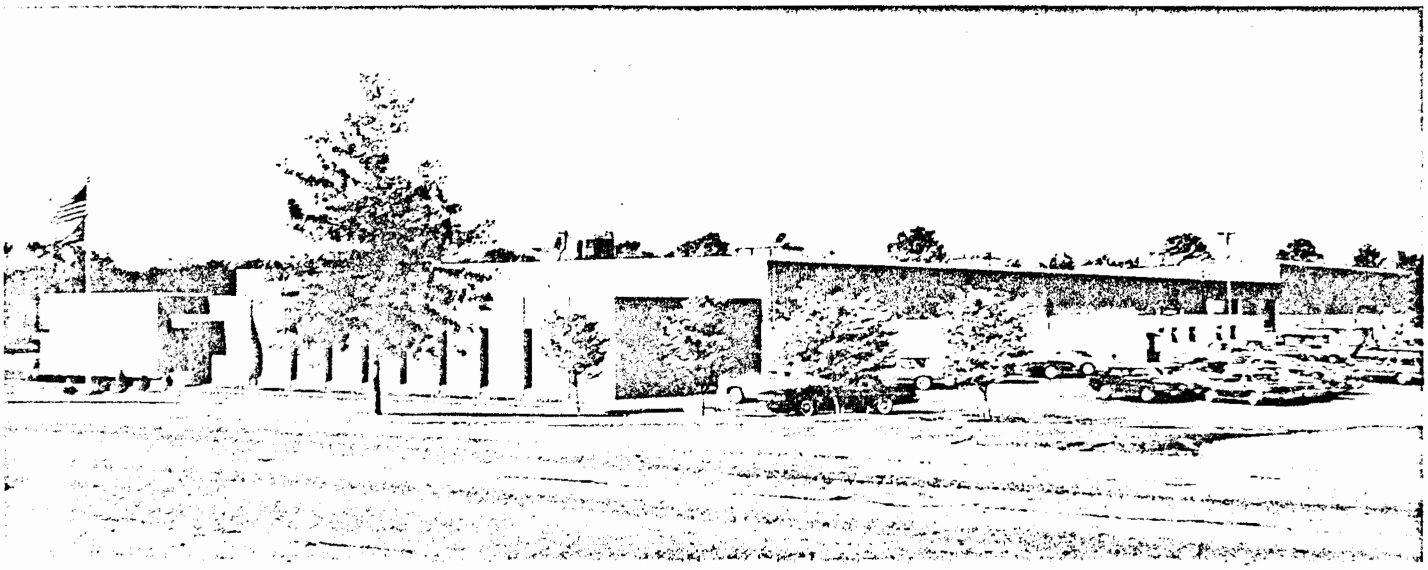




MODEL NO	NO OF RACKS	TORQUE @ 2000 P.S.I. LBS. IN	TORQUE @ 138 BAR NM	ROTATION DEGREES	DISPLACEMENT IN³	DISPLACEMENT CM³	A IN	A MM	B* IN/MM	C* IN/MM	D IN/MM	E IN/MM	F IN/MM	G IN/MM	H* IN/MM	J IN/MM	K IN (W×L)	L IN/MM	M IN (W×H)	N IN/MM	P IN	R IN/MM	T IN
H6	1	1200	135.6	100 190 280 370	1.16 2.21 3.25 4.30	19.01 36.22 53.27 70.48	8.11 10.47 12.83 15.18	205.99 265.94 325.88 385.57	N/A	4½"	4.25	.375	2.995 3.000	3.18	N/A	.996 .998	¼" × 1"	1.28	3/16" × ⅛"	.875 .878	¼" N.P.T.	3.62	5/16" NF
[illegible]	2	2500	282.5	100 190 280 370	2.32 4.42 6.50 8.60	38.02 72.44 106.54 140.96	8.11 10.47 12.83 15.18	205.99 265.94 325.88 385.57	N/A	114.30	107.95	9.53	76.07 76.20	80.77	N/A	25.30 25.35		32.51		22.23 22.30	OPTIONAL SAE ⅜"-24	91.95	× ⅝" DP
H19	1	3600	406.8	100 190 280 370	3.62 6.88 10.14 13.39	59.33 112.76 166.20 219.46	10.24 13.38 16.52 19.66	260.10 339.85 419.61 449.37	3 9/16"	5½"	5.38	.460	3.495 3.500	4.36	3¾"	1.248 1.250	5/16" × 1¼"	1.63	¼" × 3/16"	1.000 1.004	¼" N.P.T.	4.38	½" NC
[illegible]	[illegible]	7800	881.4	100 190 280 370	7.24 13.76 20.28 26.78	118.66 225.52 332.40 438.92	10.24 13.38 16.52 19.66	260.10 339.85 419.61 499.37	90.49	139.70	136.65	11.68	88.77 88.90	110.74	95.25	31.70 31.75		41.40		25.40 25.50	OPTIONAL SAE 7/16"-20	111.[illegible]	× ½" DP
[illegible]	[illegible]	12000	1356.0	100 190 280 370	12.00 22.79 33.58 44.38	196.68 373.53 550.38 727.39	12.99 17.39 21.79 26.19	329.95 441.71 553.47 665.23	5"	8"	7.75	.620	4.995 5.000	6.18	5¼"	1.998 2.000	½" × 2"	2.97	⅜" × ¼"	1.751 1.753	½" N.P.T.	6.[illegible]	⅝" NC
H133	2	25500	2881.5	100 190 280 370	24.00 45.58 67.16 88.76	393.36 747.06 1100.76 1454.78	12.99 17.39 21.79 26.19	329.95 441.71 553.47 665.23	127.00	203.20	196.85	15.75	126.87 127.00	156.97	133.35	50.75 50.80		75.44		44.48 44.53	OPTIONAL SAE ⅞"-16	155.45	× ⅞" DP
[illegible]	[illegible]	48000	5424.0	100 190 280 370	48.03 91.26 134.49 177.72	787.21 1495.75 2204.29 2912.83	24.38 33.03 41.53 49.78	619.25 838.96 1054.86 1264.41	7.00	12.00 †	12.25	.980	8.495 8.500	8.96	8.495 8.500	2.998 3.000	¾" × 2⅜"	3.02	¾" × ½"	3.001 3.003	¾" N.P.T.	10.00	¾" NC
501	2	103000	11639.0	100 190 280 370	96.06 182.52 268.98 355.44	1574.42 2991.50 4408.58 5825.66	24.38 33.03 41.53 49.78	619.25 838.96 1054.86 1264.41	177.80	304.80 †	311.15	24.89	215.77 215.90	227.58	215.77 215.90	76.15 76.20		76.71		76.23 76.28	OPTIONAL SAE 1 1/16"-12	254.00	× [illegible] DP
[illegible]	[illegible]	221000	24973.0	100 190 280 370	222.21 422.21 622.20 822.19	3642.02 6920.02 10197.36 13475.69	38.03 53.53 74.28 81.53	965.96 1359.66 1886.71 2070.86	11.00	19.00	19.00	1.220	13.995 14.000	13.45	13.955 14.000	4.998 5.000	1" × 4⅞"	6.90	1" × ¾"	4.501 4.507	[illegible] N.P.T.	17.[illegible]	[illegible] NC
[illegible]	[illegible]	478000	54014.0	100 190 280 370	444.42 844.42 1244.40 1644.38	7284.04 13840.04 20395.72 26951.39	38.03 53.53 74.28 81.53	965.96 1359.66 1886.71 2070.86	279.40	482.60	482.60	30.99	355.47 355.60	341.63	355.47 355.60	126.95 127.00		175.26		114.33 114.48	OPTIONAL SAE [illegible]	43[illegible].50	× [illegible] DP

† = 12.00 / 304.80   D/2 = [illegible] / 153.75   *Dimensions B, C and H are as cast for models H6 thru H133 only

OHIO OSCILLATOR
1707 North Main Street • Orrville, OH 44667
Phone (216) 683-5075 • Telex: 98-6402



OHIO OSCILLATOR...DEDICATED TO QUALITY, INTEGRITY AND LEADERSHIP IN THE ROTARY ACTUATOR MARKET.

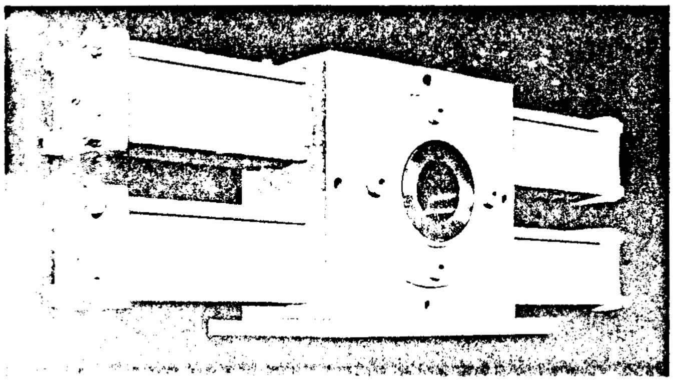
OHIO OSCILLATOR'S modern 80,000 square foot facility employs craftsmen, technicians and engineers, all dedicated to designing and building the best rotary actuators you can buy.

OHIO OSCILLATOR'S standard line of rotary actuators finds application in every market where economical, dependable, instant-high-torque power sources are required. Special configurations built to customer specifications are commonplace.

OHIO OSCILLATOR'S rotary actuators have ZERO LEAKAGE (internally) to provide pinpoint positioning and positive holding for infinite time periods.

OHIO OSCILLATOR'S proven rack and pinion design provides rotations up to 1800° (five revolutions).

OHIO OSCILLATOR, a service-oriented source you can depend on for the very best of rotary actuators today, is...
"The Force for The Future."

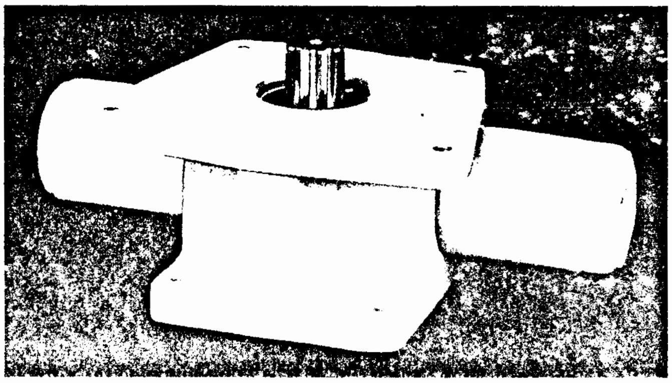


Series "HH" Rotary Actuators

[illegible] Series are Custom-Designed Units to meet Your [illegible]ations for High Torque Output Applications. (Tie Rod or [illegible]) Consult Factory for Details.

[illegible]d Design Features...
- [illegible] Pressures: 250 - 3000 PSI
- [illegible] Torques: 83,500 - 50,000,000 Inch Pounds
- [illegible]ard Rotations: 90°, 180°, 270°, 360°

[illegible]ions...
- [illegible]ators
- [illegible]ation
- [illegible] Cars
- [illegible]aring, etc.
- [illegible] Imagination Work.



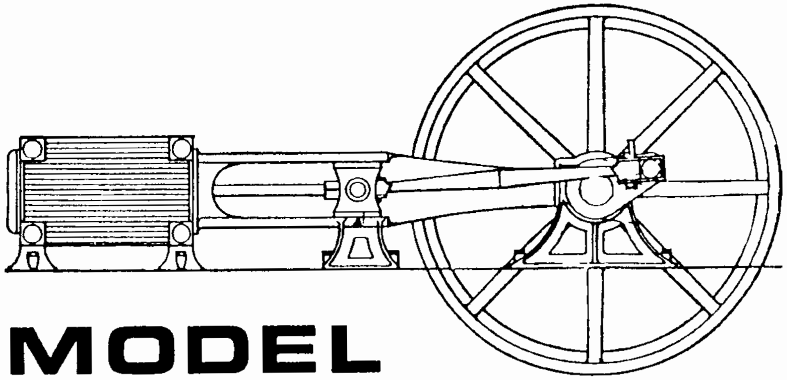
Series "M" Rotary Actuators

"M" Series are Custom-Designed Units for the Mobile Market. Consult Factory for Details.

Standard Design Features...
- Input Pressures: 100 - 2,000 PSI
- Output Torques: 1,100 - 47,800 Inch Pounds
- Standard Rotations: 90°, 180°, 360°

Applications...
- Mobile Equipment
- Construction Equipment
- Elevated Platforms

PHOTO SHOWS JUST ONE OF
MANY CONFIGURATIONS
DESIGNED TO MEET SPECIFIC
NEEDS OF THE MOBILE MARKET.



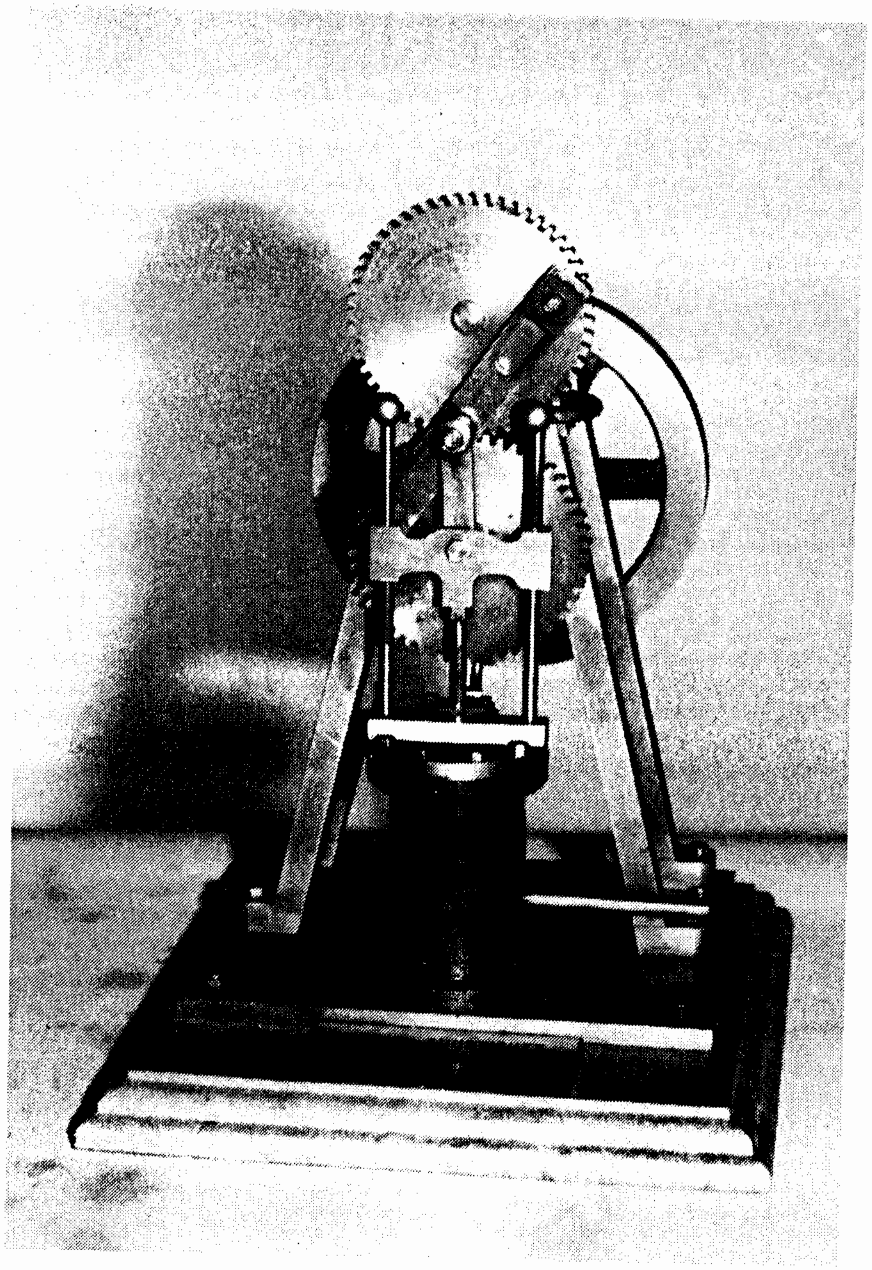
MODEL MACHINES

Replica Steam Models

MARLYN M HADLEY
29007 TAMPICO RD
CORVALLIS OR 97330

MARLYN HADLEY

Table of Contents



Crank Substitute

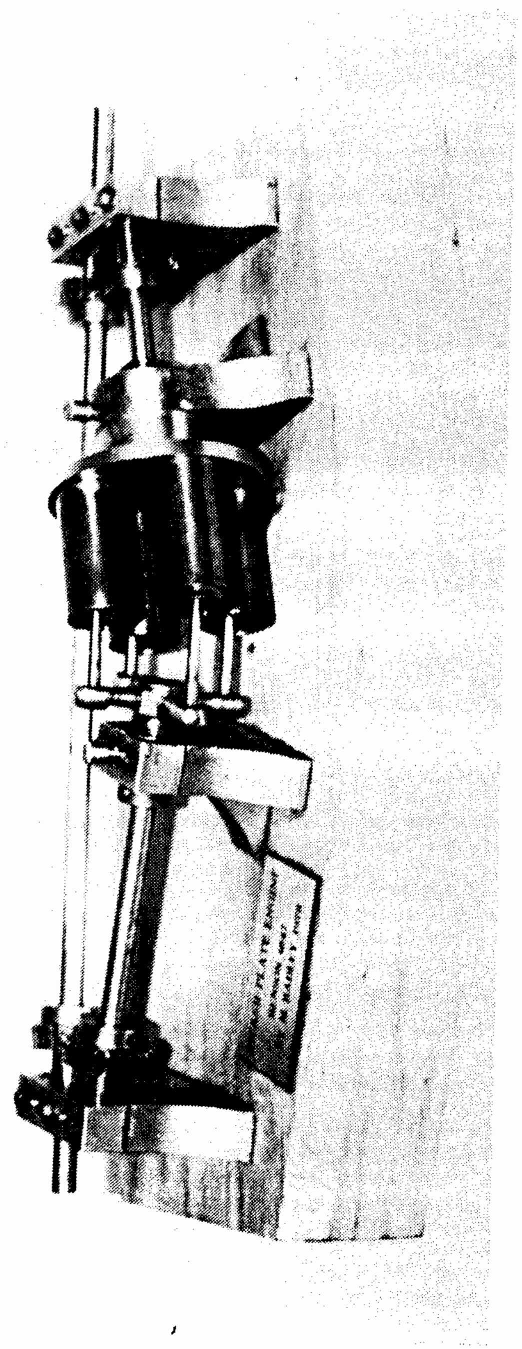
This elaborate arrangement of gears and linkages enabled the builder to eliminate the crank as we know it. While this engine required more labor to construct, it did make a compact engine which did not require a heavy crosshead as the connecting rod connection on the bar between the two gears moves but a very small amount. This means the piston rod guides can be made of a lighter construction.

I do not know who invented it, when, or what prompted the inventor to think this arrangement is better than a crank. This system was used, however, for some small water pumps.

This model has a one-inch bore and two-inch stroke using the same cylinder I have used on several models.

This model was made from a very small illustration in the book "Mechanical Appliances and Novelties of Construction" by Gardner D. Hiscox in 1904.

This model was made in 1982.



Swashplate Engine

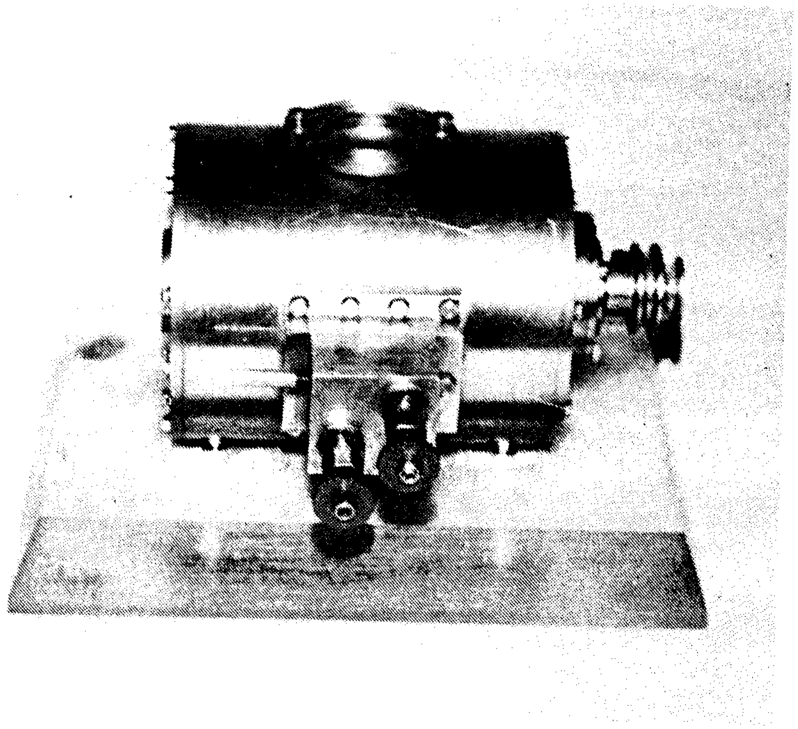
This engine was invented by Benson in 1847 to overcome some problems that are inherent in most steam engines. This engine has no dead center and so is self-starting in all positions and it does not need a heavy flywheel to steady the motion, also the turning torque in this engine is the same in any position.

In this design, four small single-acting cylinders are clustered on one shaft and the piston rods are joined to rigid arms on another shaft. As the two shafts are at a slight angle, the pistons reciprocate as the shafts turn, reaching their extremes at the "9 o'clock" and "3 o'clock" cylinder positions. The connecting rods have a ball turned on each end, seated in sockets, to allow for the changing angles. The long shaft with its bevel gears acts both to synchronize the pistons and cylinders and as the output shaft.

While it was patented as a steam engine, there is no evidence that many were made. The swashplate principle is used today in many hydraulic pumps and motors with very good success as the unit can be made very compact.

This model was made from a photograph taken by Marlyn Hadley of a patent model in the Smithsonian Institution at Washington, D.C. The model has four cylinders 13/16 inch bore and ⅞ inch stroke and the angle of the shafts is 15°. I cut the bevel gears and made the brass castings in my workshop.

This model was finished in 1978.



Tesla Turbine

Nikola Tesla is commonly thought of as an electrical inventor, but his interests were varied. We are surprised to learn of his interest in steam and many other fields. His steam turbine of 1906 is different from other turbines in that the revolving discs are smooth. The power is developed by friction of the steam on the sides of the discs. Tesla's turbine was 2 feet tall and 3 feet long and produced 200 horsepower at 16,000 revolutions per minute, at 125 pounds steam pressure. It had 25 discs of hardened steel, 1/32" thick, spaced about 1/32" apart. The steam spirals around the discs and exhausts through holes in the discs near the shaft endways to the exhaust holes at each end. The nozzle is a narrow slot which blows steam on the edge of the discs at a tangent. They claimed the exhaust pressure was 1 pound per inch. One advantage of this turbine is the ease of reversing, which is impossible with conventional turbines. They could be reversed by closing one valve and opening another, allowing the steam to be blown on the discs in the opposite direction.

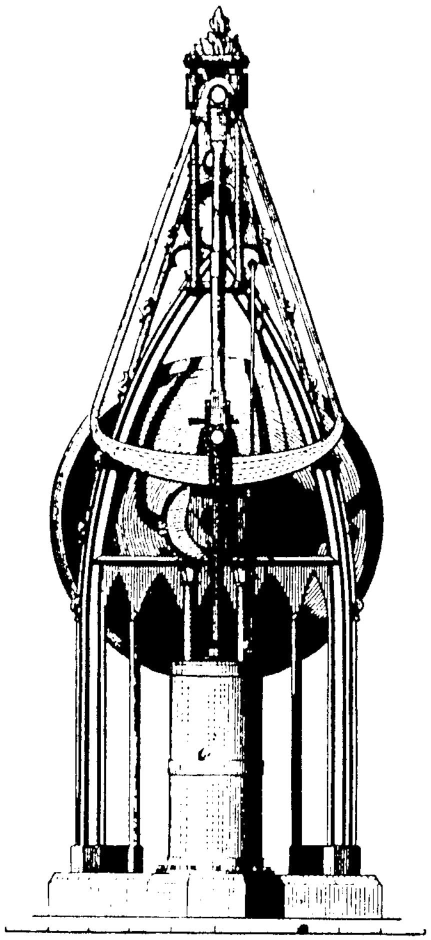
At the working speed of 35,000 revolutions a minute, the centrifugal force was so great that it stretched the metal in the discs. It was many years before better metal alloys were developed capable to stand the terrific strain of the speed of the discs.

Today interest in the Tesla turbine is rising and at last the idea is getting the attention it deserves.

Now in the 1980's this unique turbine is under development for automobile and power plant use by Sun Wind Co. of Sebastopol, California. Instead of steam this company plans to burn porpane, alcohol, gasoline or hydrogen directly into the turbine in a three-wheel car called the rainbow as one of their researchers built a turbine which performs as Tesla predicted it would. They are now working to design a proper combustion chamber.

This model of Tesla's turbine was built from an article in the September 1965 issue of "Popular Mechanics." The model has a series of stainless steel discs 2 ¾" in diameter. The nozzle is a slot ½" wide and 3/1000" thick. The slots are angled at a tangent to the edge of the discs and strike the discs about 3/16" from the outside and spiral to the exhaust holes near the shaft. I do not know the exact speed, but the best estimate with 80 pounds of air is over 10,000 R.P.M.

This model was finished in 1967.



(2)

MECHANICS

FOR

THE MILLWRIGHT, MACHINIST, ENGINEER, CIVIL ENGINEER, ARCHITECT AND STUDENT.

CONTAINING

A Clear Elementary Exposition

OF THE

PRINCIPES AND PRACTICE OF BUILDING MACHINES.

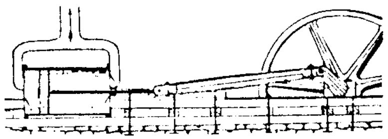
BY
FREDERICK OVERMAN,
AUTHOR OF "THE MANUFACTURE OF IRON," AND OTHER SCIENTIFIC TREATISES.

ILLUSTRATED BY ONE HUNDRED AND FIFTY-FOUR FINE WOOD-ENGRAVINGS,
BY WILLIAM GIHON.

PHILADELPHIA:
LIPPINCOTT, GRAMBO & CO,
SUCCESSORS TO GRIGG, ELLIOT & CO.
1851.

the crank is moved one-quarter of the circle, the force exerted upon the compressed air is checked, and the speed of the wheel reduced. This variable resistance causes an

Fig. 83.



irregular speed in the water-wheel, and consequent loss in power. In this case we find still more loss of power than that arising from the irregular motion of the wheel. If, at the circumference of the water-wheel, there is an active force like water, that force will be increased in the crank, should the circle which the latter describes with its pin be smaller than the first, and diminished if larger, in proportion to the lengths of the circumferences. The active force exerted by the crank is, however, not more than two diameters, in the best case; that is, if no effect of the water-wheel is lost by the irregular motion. The oscillatory motion of the piston meets in every part of its way a certain resistance, which is in this case the pressure of the blast. In the middle of the stroke, or when the crank is at right angles with the direction of the linear motion of the piston, the resistance may be assumed to be equal to the force. If we need a certain density of blast, we are under the necessity of constructing the water-wheel and crank so as to produce that density when the piston is at its highest speed, or at right angles to the piston-rod. The resistance is here regulated by the nozzles of the blast-pipes, and is as the piston's motion; and as the moving

power may be assumed to be uniform in its circular motion, it cannot increase the pressure of the blast, because the velocity of the piston is a condition necessary to increase that pressure. The latter cannot be realized; on the contrary, it diminishes gradually to the dead point of the crank, and increases from there in its return motion. We experience here a real loss in power, in the proportion of $2 \times D$ to $3{\cdot}1415 \times D$, or two diameters to the circle described by the crank-pin, irrespective of the loss by irregular speed in the water-wheel.

THE HALF-TOOTHED WHEEL.

The loss of power in the crank, in converting rotary into linear motion, is still more apparent if we compare it with the half-toothed wheel. In fig. 84 is a blast-cylinder, and

Fig. 84.

a conversion of the rotary motion represented, which is theoretically more perfect than the crank, but is very limited in practical application. The half-toothed wheel A, fastened to the axle of the water-wheel, will transfer the whole force of that wheel to the two racks B and C, and the piston-rod. Here, two lengths of the stroke are equal to the periphery of the cog-wheel and its velocity; consequently, no power is lost. If in both cases, that of the crank and this wheel, we have an equal power in the water-wheel, the pressure of the blast will be remarkably

greater in the latter than in the first case. If the force in the buckets of the water-wheel is equal to 1000 pounds, and if the way travelled by the piston of the blast-cylinder is to the way travelled by a point in the circumference of the wheel as one to ten, the pressure of the piston upon the air before it is ten times as great, or 10,000 pounds, minus friction and other losses arising from dead space. If we apply the half-toothed wheel for producing the linear motion, this is correct; but not in the crank. If the water-wheel is eighty feet long in its periphery, it requires four feet stroke, or eight feet motion of the piston, to make one-tenth of the way of the water-wheel. This length of stroke may be produced by a half-toothed wheel of eight feet circumference, or a radius of $\frac{8}{2 \cdot 314} = 1{\cdot}27$ feet. To impart that motion to the piston, it requires a crank of two feet; and as the pressures of the piston upon the confined air are inversely as the radius of the water-wheel to the radius which produces the linear motion, the pressure of the piston moved by a crank will in this case not amount to more than $\frac{10{,}000 \times 1{\cdot}27}{2} = 6350$ pounds, instead of 10,000 pounds. And if the surfaces of the pistons are in both cases the same, the densities of the compressed air will be as these numbers.

If we consider the connecting-rod, in this instance, which belongs to the crank, the result is still inferior to the above, for the oblique action of this rod is here disadvantageous. If there is no loss of power in the crank by converting the linear into a rotary motion, there is certainly a loss in converting the rotary into linear, in most cases.

The form of the half-toothed wheel, in converting motion, is perfect in all instances, but its application is very

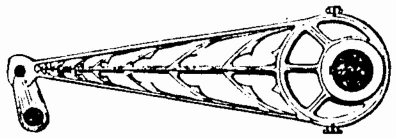
limited. At each end of the oscillating linear motion there is a moment of disconnection between the driving power and the driven machinery; this causes an increase of the velocity of the driving and a decrease in the resisting power, which causes a sudden shock or concussion between the parts of the machinery. This concussion is destructive to the machine, in case it is violent, that is, if the motions are rapid: weight, and particularly velocity, are here to be avoided. A practical speed for a blast-cylinder may be one, but not more than two revolutions per minute; no cast-iron machine will resist more than that speed. The durability of a machine, in this instance, may be greatly increased by making the time of disconnection as short as possible.

The loss of power in the crank may be obviated, in this and similar cases, by multiplying the number of cranks. Two cranks are qualified to impart more power than one, and three will yield still better. The laws developed above apply but to one crank.

THE ECCENTRIC

Is another means of converting a circular into a linear motion. The common eccentric (fig. 85) is generally ap-

Fig. 85.



plied where a crank is not practicable; it performs the same motion, and is subject to the same laws. As a me-

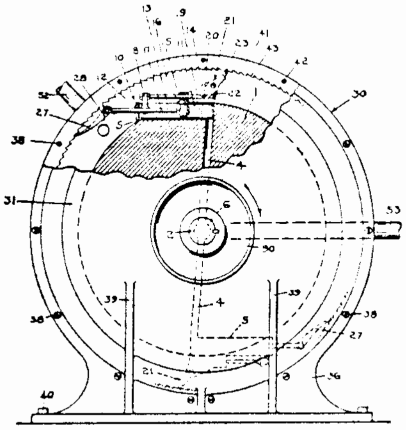
22

U. S. PATENT OFFICE

JUNE 17, 1930

1,764,324. AIR MOTOR. JAMES LIVINGSTONE, Seattle, Wash., assignor of one-tenth to Harry E. Stanley, Seattle, Wash. Filed Sept. 28, 1929. Serial No. 395,858. 6 Claims. (Cl. 121—58.)

1. An air motor, comprising a substantially airtight casing, a wheel rotatable in the casing, an annular ratchet member mounted in the casing and encircling the wheel, a piston cyclinder mounted on the wheel, a piston mounted in the cyclinder, a pawl pivotally mounted and linked with the piston, said pawl being engageable with said member, a second pawl pivotally connected with the piston and engageable with said member, and means for reciprocating the piston in said cyclinder by exhausting air within the motor and at one end of the piston and introducing atmospheric pressure within the motor and at the other end



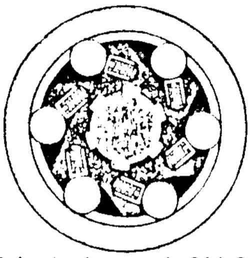
of the piston, and alternately reversing said exhaustion of air and said introduction of atmospheric pressure, and whereby a thrust is imparted to each of said pawls alternately as each is brought into operative engagement with said member by the piston.

MOTOR TREND MAY 1981

FreeWheeling

SPEAKING OF FREEWHEELING

If you thought freewheeling was one of the Fabulous Furry Freak Brothers, read on: Like so many other bits of automotive technology, freewheeling came to cars from bicycles. When the transition was made from *ped* to pedal power, a device was necessary to allow coasting with stationary



pedals. And so, each Old Ordinary (bicycle) had, as part of its specification, the one-way ratcheting device, sometimes called a sprag clutch and finally, freewheeling.

Its application to the automobile was a natural. With freewheeling, relaxation of power meant the vehicle coasted just as if a regular friction clutch had been disengaged. What actually happened was a cessation of wedging action–spring-loaded rollers that engaged a ramped pawl were free, while coasting, to roll over the ramp. On reapplication of power, the rollers remained stationary in relation to the pawls, and the whole became a driving member.

Since, in effect, a clutch was being disengaged, "clutchless" shifts up and down were possible. Combined with an "electric hand" or other rudimentary auto-shift mechanism, freewheeling was the key to the automatic transmission.

In conjunction with manual shift gearboxes, besides the possibility of clutchless shifting, freewheeling increased gas mileage by allowing the engine to return immediately to idle as the accelerator was released. Since the car was effectively coasting with the clutch disengaged, there was no compression braking (and no fuel pulled through carb circuits by high manifold vacuum). Cruising in your DeSoto, Plymouth, Buick, Rover, Nash, or Austin was more quiet, economical and relaxed.

Unfortunately, brakes of the period needed all the help available, and while the freewheeling could be locked out, lockout was difficult while the car was moving. By WWII, economics, overdrive and automatic transmissions (both of the latter use freewheeling as an integral part) made the simple freewheel redundant.

It was revived by Saab for the company's 2-strokes (and early 4-stroke V-4), but after that it remained on the shelf . . . until recently. Gas mileage is again king, disc brakes make stopping more a certainty, low car weight further reduces braking loads, and long gearing with small engines reduces engine braking effect anyway.

The resurgence of the manual transmission, too, makes the advantages of freewheeling apparent. Again: clutchless shifting once the car is moving, reduction in engine wear, increase in gas mileage.

—*Len Frank*

Bring Back Freewheeling

MY TURN/JACK AVINS

Newsweek

We could reduce our gasoline consumption by as much as 15 per cent if we were to provide incentives for the auto industry to revive freewheeling, which was widely used in the early 1930s. The savings would approach 1 million barrels of oil daily and an annual reduction in our oil bill of some $10 billion.

Freewheeling eliminates the waste of energy that occurs when engine drag in a conventional car prevents it from coasting freely and thereby using the energy stored in the momentum of the vehicle. When properly implemented, freewheeling does not compromise safety: the driver retains complete power to brake or accelerate the car.

In bicycles, freewheeling makes it unnecessary to keep pumping the pedals when there is enough momentum to carry the rider at the speed he wishes to travel. Freewheeling is thus familiar to anyone who has ever ridden a bike, whether of the single-speed or the ten-speed variety.

For the past 45 years, the auto industry has designed its cars so that, unless gas is continually fed to the engine, the engine will slow down the car under conditions where it would otherwise continue to coast freely. This engine drag is greatest in cars with manual transmissions and least at cruising speeds in cars having automatic transmissions. A freewheeling device eliminates this drag without adding significantly to the cost of the vehicle.

The story of why we have continued to build cars without freewheeling provides a fascinating chapter in automotive history.

In 1932 virtually every auto manufacturer used freewheeling, including Chevrolet, Oldsmobile, Lincoln and Chrysler, as well as less-familiar names since vanished from the scene: Auburn, Dusenberg, Franklin, Hupmobile, Marmon, Pierce Arrow, Studebaker and Willys-Knight. The foremost exponent and pioneer in freewheeling was a distinguished engineer, D. G. Roos, a president of the Society of Automotive Engineers and the chief engineer of the Studebaker Corp.

The primary reason for freewheeling was the desire to save fuel. The industry's experience during this period established beyond any doubt that freewheeling reduced fuel consumption by more than 15 per cent.

The safety issue was addressed head-on from the start. In the Journal of the Society of Automotive Engineers for January 1931, Roos reported that, before production began, "cars equipped with freewheel transmissions were submitted to many state motor-vehicle officials for trial and comment. Without exception the approval was unanimous and enthusiastic."

Although freewheeling was demonstrably as safe as conventional transmissions because of the ease of engaging engine braking in a lower gear, the safety issue continued to be a negative psychological factor in the eyes of the public. In addition, gasoline was then cheap (about 20 cents a gallon) and environmental pollution was not a problem. For these reasons, together with the fact that the auto industry concluded it could save money by dropping freewheeling, by 1935 freewheeling had virtually disappeared.

A number of recent developments suggest that the revival of freewheeling would be in the public interest:

■ The reduction in fuel consumption made possible by freewheeling and related improvements is now critical to our national defense and our economic stability. Because we have to import so much of the oil that we consume, we are highly vulnerable to any stoppage of the flow of oil from the Middle East. And even if there should be no such interruption, we clearly cannot continue to pay the cartel-imposed price without adding substantially to inflation.

■ Freewheeling opens the door to further reductions in fuel consumption. In combination with an electronically controlled gas pedal that automatically closes the throttle when engine torque is not required, without the driver having to remove his foot from the accelerator pedal, freewheeling accomplishes even greater fuel savings. Additional savings are possible by designing the vehicle so that the engine can be safely turned off under certain conditions and by providing for automatic restarting when the accelerator pedal is depressed.

■ Improved techniques are available for making freewheeling autos as safe as the cars we now drive. In particular, the normal reflex of stepping on the brake pedal can be made to automatically provide full engine braking. Freewheeling is restored when the accelerator pedal is depressed.

An old-fashioned device can help us to reduce gas consumption inexpensively and safely.

To determine whether freewheeling is in the public interest, an industry-government committee should be set up. If the decision is in the affirmative—and the evidence points overwhelmingly in this direction—new EPA mileage-rating schedules should be designed to evaluate and give credit for the fuel savings realized under actual driving conditions. (Ultimately, these are the ratings that appear on the windows of new cars.) Without usurping any of the prerogatives of the auto industry or adding to government regulations, this would provide the needed incentive for manufacturers to adopt freewheeling and related improvements. The need is urgent and the program should be implemented without delay.

Avins, a fellow of the Institute of Electrical and Electronics Engineers, recently presented his ideas on fuel conservation to the Department of Transportation.

RATCHETS AND OVER-RUNNING CLUTCHES

As every boy who has dissected an old alarm clock knows, the familiar ratchet on wind-up mechanisms is a sort of biased gear, the teeth having a sharp slope on one side and a more gradual one on the other. Held against the teeth is a pawl or detent (Figure 17) that catches positively against the nearly radial slope so that ratchet and pawl are locked against each other in one direction. But if the ratchet is turned the other way (or the pawl in the opposite direction) it is allowed to more as the pawl climbs the longer slope.

On a railroad brake or a winch, the pawl may be pivoted on a solid member so as to engage of its own weight, and can be flipped out of the way when not wanted. In a spring-wind mechanism it is usually mounted on another rotating member and requires a light spring to keep it engaged at all times.

Such a ratchet clicks or chatters when running free. It can be fitted with a friction collar and lever that lift the pawl away from the teeth in the free-running direction. A quiet ratchet that will not only lock anywhere the wheel stops (instead of only in tooth increments) is shown in Figure 18. The friction pawl may be weight or spring loaded, and it is sometimes fitted with a loose shoe or pad to increase the contact area and lessen wear.

Another stepless ratchet makes use of torsional friction. In Figure 19, the shaft is to turn freely counterclockwise when seen from the right-hand end, but lock when turned in the other direction. Fixed to the shaft is a small barrel, and wound around this is a very close-fitting coil spring. One end of the spring simply lies against the barrel, but is not fastened to it. The other end is hooked around a fixed pin or otherwise firmly anchored.

The shaft being turned counterclockwise, friction between the close-hugging coils of the spring and the barrel will tend to open or unwind the coils. This allows the shaft to turn freely. But if the shaft is turned clockwise, friction at once tightens the coils around the barrel, wrapping them so firmly that the shaft cannot be turned.

An over-running clutch in an automobile overdrive resembles a roller bearing, but with one important difference. The outer race, which has a smooth concentric bore, is the driven member (Figure 20). The inner race is the driving member, but its outer surface is not circular. Instead, it has a

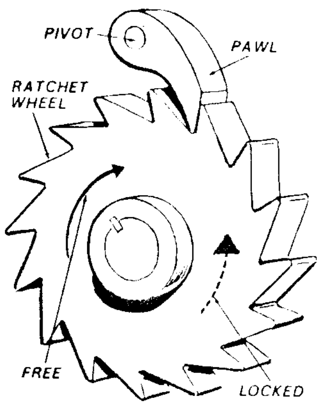


Fig. 17. Ratchet and pawl mechanism.

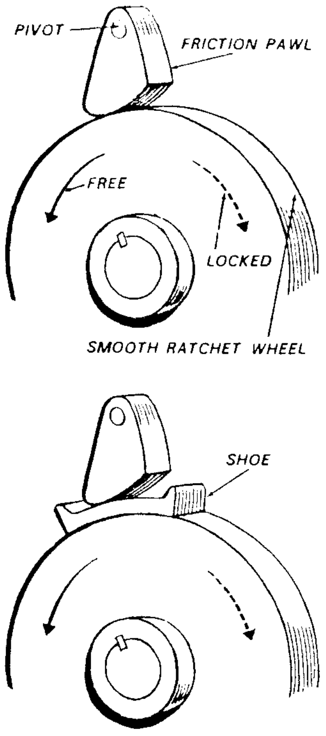


Fig. 18. Quiet ratchet works by friction. Pawl may be fitted with a shoe as in bottom drawing.

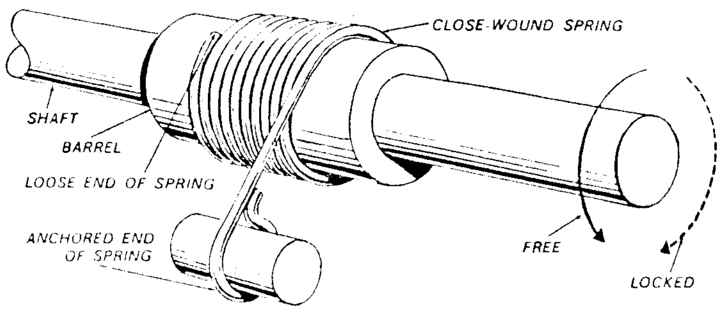


Fig. 19. Stepless ratchet makes use of torsional friction.

number of evenly spaced sloping flats or cam ramps machined on it. Between the inside of the outer race and each of these ramps is a roller.

When driving torque is applied to the inner or cam race, it moves inside the rollers, wedging them between the ramps and the outer race so firmly that the whole clutch revolves as a unit. Thus torque is transmitted from the inner to the outer race.

If you lift your foot from the throttle, letting the engine idle while the car coasts, the outer race will overspeed the inner one. This rolls the rollers down to the lower part of the cam ramps, where they may revolve without wedging. The outer race therefore free-wheels independently of the inner one. But if the engine is revved up again, the inner race overtakes the outer one, the rollers are forced up the ramps, and once more wedge the outer race tightly.

Another rack mechanism has two rows of teeth on a single member (Figure 27). The pinion has teeth on only half its circumference. As in turning clockwise the pinion meshes with the lower teeth, it moves the rack to the left. After half a turn, the pinion teeth mesh with the upper row, moving the rack to the right.

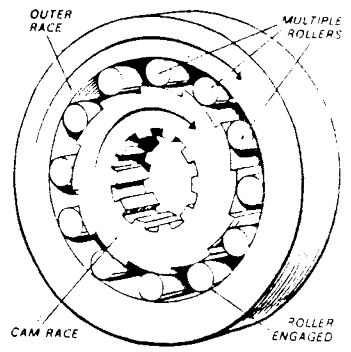


Fig. 20. Over-running clutch.

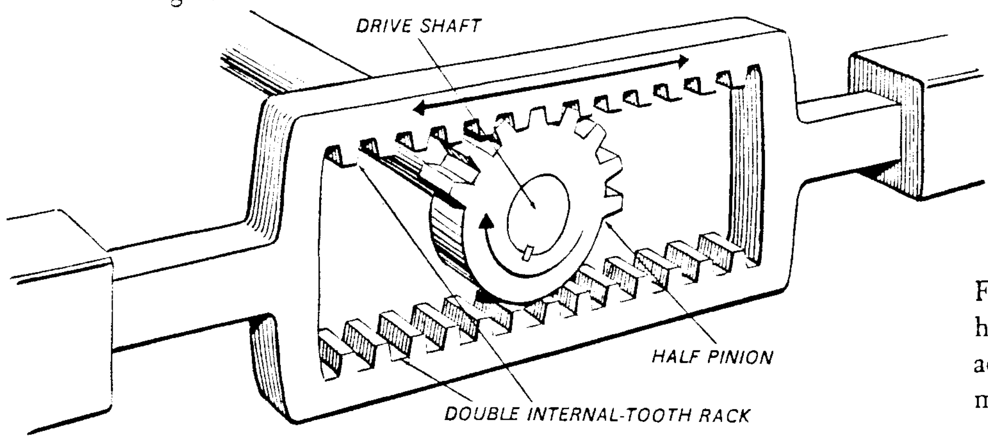


Fig. 27. Rotary motion of half pinion on drive shaft activates reciprocating motion in double rack.

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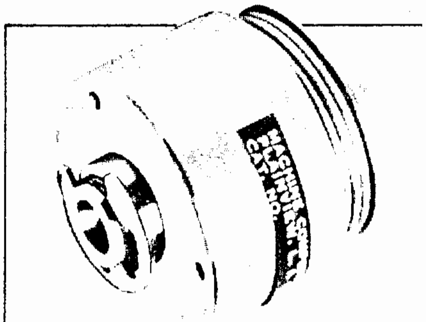
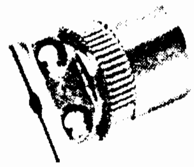


GM-B series clutches are precision calibrated, spring-wrapped slip clutches. They are designed for long life under continuous slip. Slip torque is stable and independent of velocity, including acceleration from rest.

Standard slip torques range from 1 — 480 oz.-in., bores from 1/8" to 1/2" dia. Slip is bidirectional. Torque values for CW and CCW directions are independent of each other and may be specified the same or different for the two directions. Provisions are made on clutches for mounting an input gear, sprocket or pulley. Units are made of hardened stainless steel.

Miniature Geared Clutches



These precision slip clutches maintain all of the features found in the standard GM-B series clutches with the addition of an integral spur gear as part of either the input or output drum.

All stock gear drums are phosphor bronze, AGMA quality 10, 20° P.A. Available from 48 to 120 D.P. Standard tolerance on pitch diameter is minus .001". Other materials, classes, and pressure angles are also available.

Standard bores range from 3/32" to 1/4" dia. Torque range is from 0.5 to 20 oz.-in. Choice of jaw clamp hub (shown) or standard set screw hub. Special gears and torques are available.

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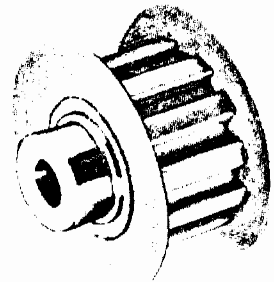
Toothed Pulley Clutches



This series offers the same features found in the GM-B series of slip clutches with the additional feature of an integral 1/5" pitch gearbelt pulley. Torques range from 1 to 480 oz in., bore diameters from 1/8" to 1/2". Pulleys are available with from 10-48 teeth. Special 3/8" or 1/2" pitch pulleys are also available.

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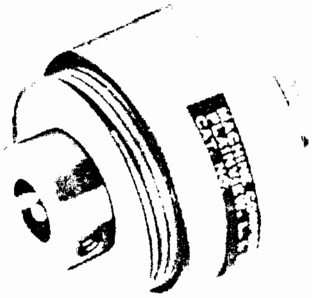
slip couplings

C, CL and CM Series



[illegible] couplings serve as torque limiters as well as couplings for two colinear shafts. When the [illegible] exceeds the limit torque, the two shafts rotate relative to each other at the full limit torque. [illegible] couplings are designed to operate with axial misalignments of up to .010" between the two shafts. Standard angular misalignment is 1.5° max. Bore diameters can differ for the two ends so that different diameters of "in-line" shafts can be coupled together.

These couplings maintain all the features of our standard line of GM-B slip clutches. Torques range from 1 oz.-in. to 125 in.-lb., bore diameters from 1/8" to 7/8". Setscrews or clamp type hubs are standard.

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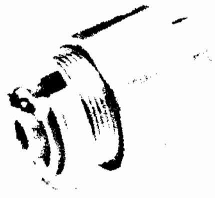
drag brakes

[illegible]-B-Series



[illegible]-B series drag brakes are similar to series GM-B slip clutches with the addition of a slotted keyway in the output drum. This slot provides a means for locking the output drum to a stationary structure making the unit functional. Torques range from 1 to 480 oz.-in., bores from 1/8" to 1/2" dia. Different values of clockwise and counter clockwise torques may be specified.

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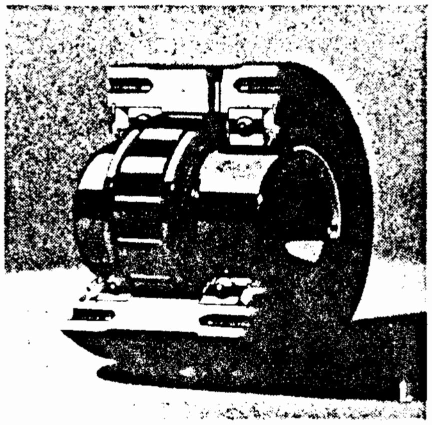

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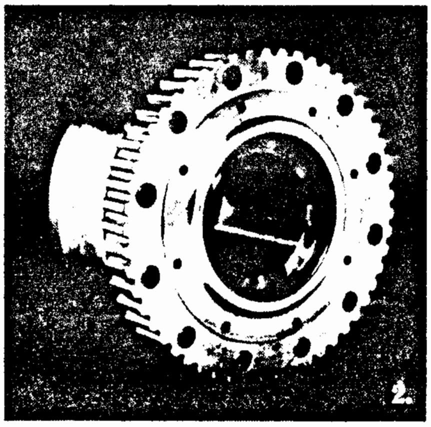

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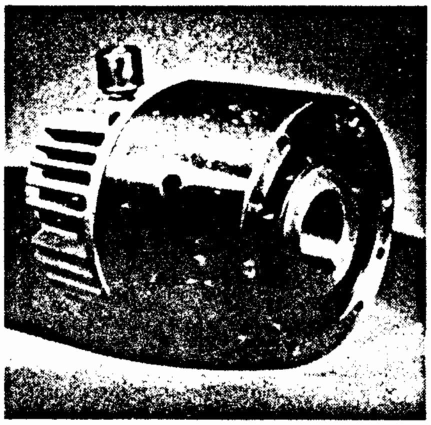

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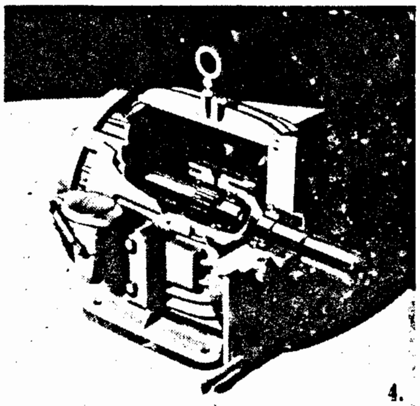

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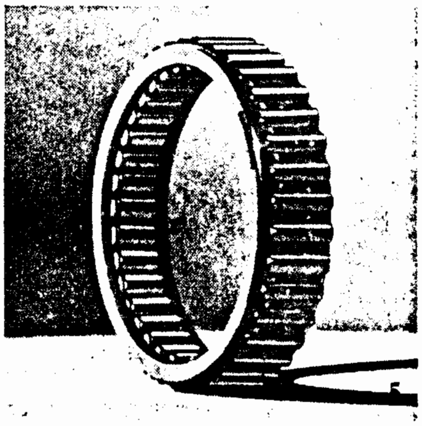

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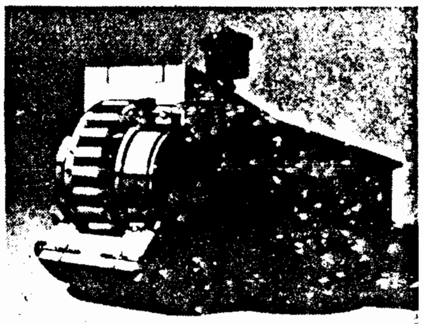

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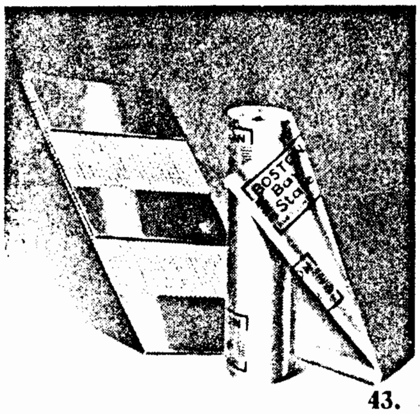

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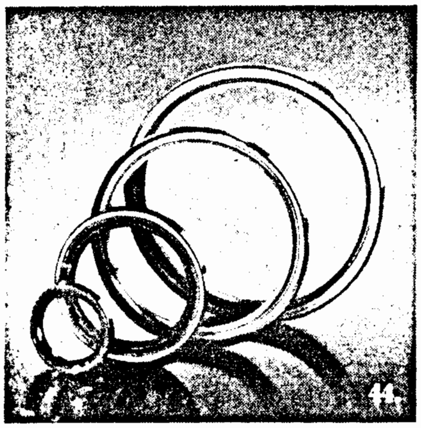

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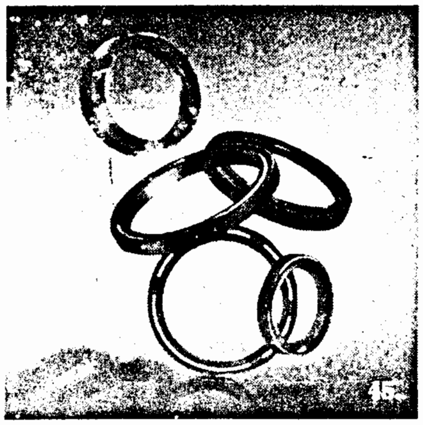

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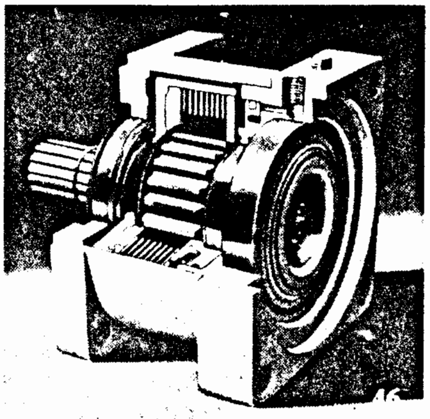

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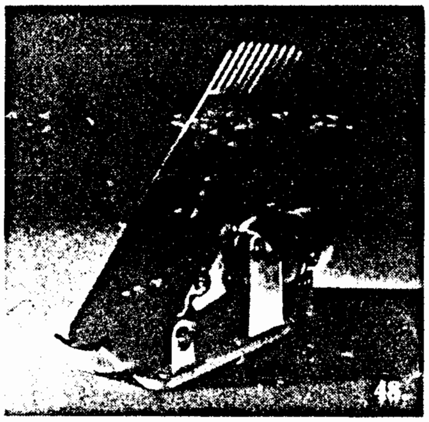

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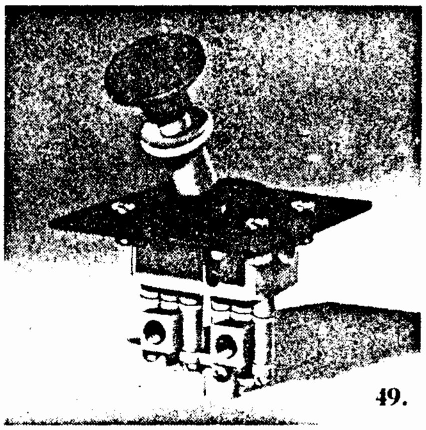

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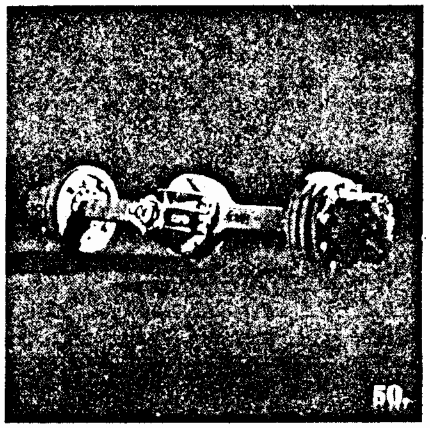

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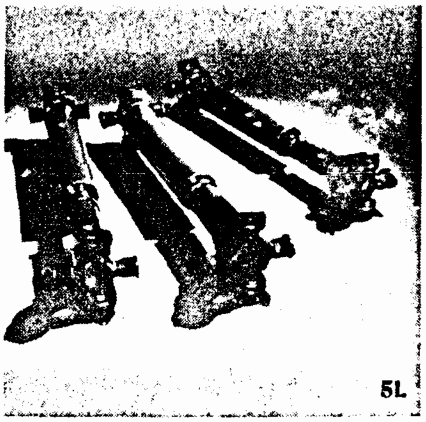

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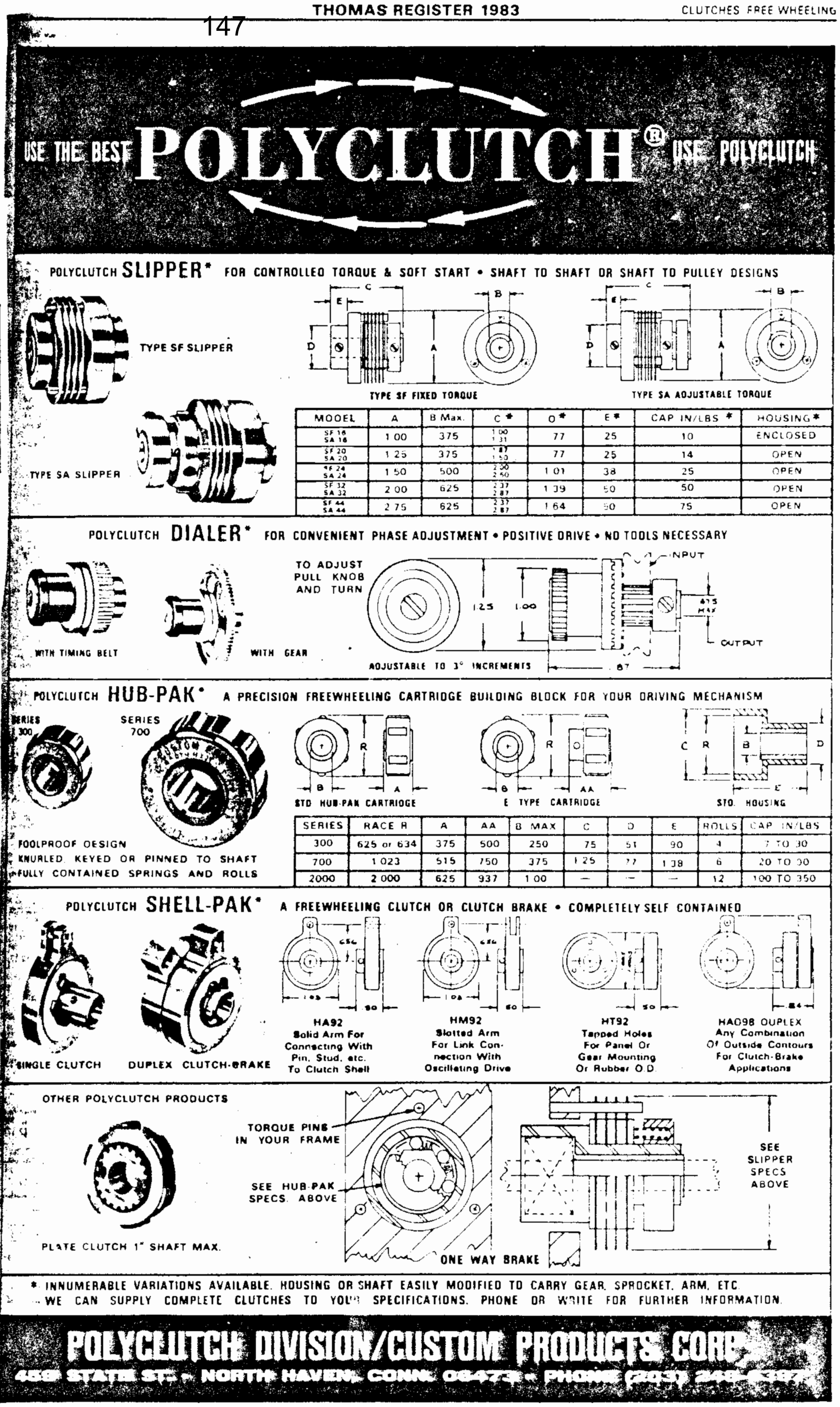
Equipment Concepts M.P.O. Box 2135-T .... NR

OHIO: TOLEDO
DANA CORP. 4500 Dorr St. (ZIP 43697) (Out Of State Call Toll Free: 800-343-3027) .... 50M+ (See Our Corporate Advertisement In Vol. 10)
***See our catalog in* THOMCAT vols. 12-17**
OKLA: ENID
V-Flex Corp. P.O. Box 762 .... 1/4M-
SC: FOUNTAIN INN
Marquette Metal Products Co. 400 S. Main St., P.O. Box 398 .... 1M+
WIS: MILWAUKEE
PT COMPONENTS, INC., STEARNS DIVISION 120 N. Broadway (ZIP 53202) (414—272-1100) .... 10M+ (See Our Corporate Profile In Volume 11)
WIS: RACINE
Twin Disc, Incorporated 1340 Racine St. .... 5M+

CLUTCHES: OVER-RUNNING

(see also Clutches: Automobile, Truck, Bus, etc.)

CAL: SANTA FE SPRINGS
~~Mueller, Georg, Of America, Inc.~~, Western Division 11823-T S. Siauson Ave. 24 .... 50M+
CONN: BROOKFIELD
Mueller, Georg. Of America, Inc., East Coast 304-T Federal Rd .... 50M+
CONN: CLINTON
HELANDER PRODUCTS, INC. 30 Knollwood Dr. (ZIP 06413) (Miniature Single Revolution Clutches & Miniature Overrunning Roller Clutches For Business Machines, Control Devices, Instruments, Film & Paper Drives, Spring Clutches) (203—669-7953) .... 1/4M+
(See Our Company Profile In Volume 10)
CONN: HAMDEN
HIGH PRECISION, INC. 379 Morse St., P.O. Box 6157 (ZIP 06517) (Miniature) (203—777-5395) .... 1/2M+
CONN: NORTH HAVEN
CUSTOM PRODUCTS CORP. 459 State St. (ZIP 06473) (Motion Control Devices For 1/2 To 1 Inch Shaft, Slip Clutches With Precision Torque Control, One Way Or Bi-Directional Brakes, Free Wheeling Roller Clutches, Single Or Partial Revolutions Clutches, Phase Adjustive Device) (203—248-6397) .... 1/2M+
CONN: TORRINGTON
TORRINGTON CO., THE, BEARINGS DIVISION 59 Field St. (ZIP 06790) (203—482-9511) .... 50M+ (See Our Company Profile in Volume 11)
***See our catalog in* THOMCAT vols. 12-17**
FLA: WINTER HAVEN
American Vulkan Corporation P.O. Drawer 673 .... 1M+
ILL: LA GRANGE
Marland One Way Clutch Div., Zurn Industries Inc. P.O. Box 308, Dept. T .... 60M+
ILL: SCHAUMBURG
MUELLER, GEORG, OF AMERICA, INC. 715-D Estes Ave. (ZIP 60193) (312—893-3334) .... 50M+ (See Our Company Profile in Volume 10)
ILL: SOUTH BELOIT
WARNER ELECTRIC BRAKE & CLUTCH COMPANY 449 Gardner St. (ZIP 61080) (815—389-3771) .... 1M+ (See Our Company Profile in Volume 11)
KAN: WICHITA
Carlson Co., Inc. The 6045 N. Broadway (Air) .... 1M+
MASS: WORCESTER
LOWELL CORPORATION P.O. Box 158, 97 Temple St. (ZIP 01613) (Ratchet & Roller, Overrun & Index) (617—756-5103) .... 1M+
MICH: WARREN
Industrial Power Transmission Div., Dana Corp. 23601 Hoover Rd., P.O. Box 778 .... 1M+
INDUSTRIAL POWER TRANSMISSION DIV. DANA CORP. 23601 Hoover Rd. (ZIP 48090) (Out Of State Call Toll Free: 800-521-3788) .... 50M+ (See our 12 page Corporate Profile in Vol. 10 under Dana Corp.)
MINN: MINNEAPOLIS
... AND RESEARCH & ENGINEERING, INC. 2420 ...a Lane, MS-1A (ZIP 55441) (612—475- ...) .... 1M+
MINN: ST. PAUL
... PRECISION MANUFACTURING CORP. 1259 ... Rd. (ZIP 55110) (Custom Mechanical & Electromagnetic Wrap Spring Clutches) (612—484-...) .... 1M+



MODEL	A	B Max.	C*	D*	E*	CAP IN/LBS*	HOUSING*
SF 16 SA 16	1 00	375	1 00 1 31	77	25	10	ENCLOSED
SF 20 SA 20	1 25	375	1 87 1 50	77	25	14	OPEN
SF 24 SA 24	1 50	500	2 00 2 50	1 01	38	25	OPEN
SF 32 SA 32	2 00	625	2 37 2 87	1 39	50	50	OPEN
SF 44 SA 44	2 75	625	2 37 2 87	1 64	50	75	OPEN

SERIES	RACE R	A	AA	B MAX	C	D	E	ROLLS	CAP IN/LBS
300	625 or 634	375	500	250	75	51	90	4	7 TO 30
700	1 023	515	750	375	1 25	77	1 38	6	20 TO 90
2000	2 000	625	937	1 00	—	—	—	12	100 TO 350

MINN: WEST DULUTH
H & H Exports, Inc. 1203 93rd Ave. .... 1M+
MO: JOPLIN
Autotronics, Inc. 2700 Davis Blvd. (Electro-Magnetic Sub-Miniature) .... 1M+
NJ: PITMAN
PSI DIV., WARNER ELECTRIC BRAKE & CLUTCH COMPANY P.O. Box 118 (ZIP 08071) (609—589-0815) .... 1M+
NY: BUFFALO
Deltran Div. American Precision Industries Inc. 440-T Genesee St .... 1M+
NY: EAST ROCKAWAY
Berg, Winfred M., Inc. 511 Ocean Ave. (Precision, Adjustable) .... 1M+
NY: ELMIRA
HILLIARD CORPORATION, THE, CLUTCH DIVISION 100 W. 4th St. (ZIP 14902) (607—733-7121) .... 1M+ (See Our Company Profile in Volume 10)
NY: FARMINGDALE
General Chain Sales Corp. 27A Sarah Dr. (Complete Line Of Power Transmission Equip.) .... 1M+
NY: JAMAICA
Euromatic Equipment Co. 161-15T Rockaway Blvd. .... NR
NY: MINEOLA
Gengarelly Ted Industries 178-A Herricks Rd. .... 1/2M+
NY: NEW HYDE PARK
Perfect Gear and Instrument Div., Designatronics Corp. 55 S. Denton Ave. .... 10M+
Sterling Instrument, Div. Designatronics Inc. 55-T S. Denton Ave. .... 10M+
Stock Drive Products, Inc. A Div. of Designatronics, Inc. 55-T S. Denton Ave .... 10M+
NY: PLAINVIEW
MACHINE COMPONENTS CORP. 70-T Newtown Rd. (ZIP 11803) (516—694-7222) .... 5M+
***See our catalog in* THOMCAT vols. 12-17**
NY: SYOSSET
American Precision Products, Inc. 575-A Underhill Blvd., P.O. Box Q (Complete Range Of AN, MS, NAS, Standards & Specials) .... 1M+

CLUTCHES: OVER-RUNNING (Contd)

NY: SYRACUSE
ROLLWAY BEARING P.O. Box 4827-A (ZIP 13221) (315—457-6211) .... 50M+
(See Our Full Page Ad At Bearings: Roller, General Line)

NY: WHITESTONE
Tyco Fastening Products, Inc. 19-07-B Clintonville St. 1/4M–

OHIO: CLEVELAND
EATON CORP., AIRFLEX DIV. 9919 Clinton Rd. (ZIP 44144) (216-281-2211; For The Name Of Your Nearest Eaton Airflex Representative Call Toll Free 800-243-8160 Ext. 21; In Conn. Call 800-842-0225 Ext. 21) .... 1M+
EATON CORP., DISTRIBUTOR PRODUCTS 9921 Clinton Rd. (ZIP 44144) ((216) 523-5000; For The Name Of Your Nearest Eaton Power Transmission Distributor Call Toll Free (800) 243-8160, Ext. 7; In CT. Call (800) 842-0225, Ext. 7) .... 1M+

OHIO: DAYTON
Dayton Superior Products Inc. 400-T Linden Ave 1/4M–

Equipment Concepts M.P.O. Box 2135-T .... NR

OHIO: TOLEDO
DANA CORP. 4500 Dorr St. (ZIP 43697) (Out Of State Call Toll Free: 800-343-3027) .... 50M+
(See Our Corporate Advertisement In Vol. 10)
♦ *See our catalog in* THOMCAT vols. 12-17

PA: PHILADELPHIA
Rowland Co./ Industrial Div. 3030 N. 20th St. (Fabricators & Distributors Of Heavy Duty Industrial Parts) .... 1M+

PA: WEST PITTSTON
Renold Inc., Power Transmission Div. P.O. Box 27-T .... 50M+

SC: FOUNTAIN INN
Marquette Metal Products Co. 400 S. Main St., P.O. Box 398 .... 1M+

WIS: WEST BEND
WEASLER ENGINEERING, INC. P.O. Box 558-T (ZIP 53095) (414—338-2161) .... 1M+

CLUTCHES: PNEUMATIC

CONN: MANCHESTER
CARLYLE JOHNSON MACHINE CO., THE Main & Chapel Sts. (ZIP 06040) (203—643-1531) .... 5M+
(See Our Full Page Ad Under Clutches: Machine Tool.)

FLA: WINTER HAVEN
American Vulkan Corporation P.O. Drawer 673 .. 5M+

ILL: MOUNT PROSPECT
MATRIX ENGINEERING LTD. 2020 S. Carboy Rd. (ZIP 60056) (Stationary Or Rotating Field) (312—437-6143) .... 1/4M+

ILL: NORTHBROOK
DEUBLIN COMPANY 1915 Stanley St. (ZIP 60062) (312—272-8080) .... 10M+
(Rotating joints for pneumatic clutches.)
(See Our Company Profile in Volume 10)
♦ *See our catalog in* THOMCAT vols. 12-17

KAN: WICHITA
Carlson Co., Inc. The 6045 N. Broadway .... 1M+

ME: PORTLAND
MONTALVO CORP., THE 200T Riverside Industrial Park (ZIP 04103) (207—797-8710) .... 1/2M+

MICH: WARREN
Industrial Power Transmission Div. Dana Corp. 23601 Hoover Rd. .... 50M+

MINN: MINNEAPOLIS
HORTON MFG. CO., INC. 1181 Fifteenth Ave., S.E. (ZIP 55414) (Brake) (612—331-5931) .... 1M+
(See adv. page 2942)

TOL-O-MATIC, INC. 3rd St. at 11th Ave. (ZIP 55415) (612—333-8605) .... 5M+
♦ *See our catalog in* THOMCAT vols. 12-17
(See adv. page 2942)

NJ: PITMAN
PSI DIV., WARNER ELECTRIC BRAKE & CLUTCH COMPANY P.O. Box 118 (ZIP 08071) (609—589-0815) .... 1M+

NY: JAMAICA
Euromatic Equipment Co. 161-15T Rockaway Blvd. .... NR

NC: CHARLOTTE
CARY MANUFACTURING CORPORATION 1321 Romany Rd. (ZIP 28204) (Rotary Joints For) (704—372-6396) .... NR

OHIO: AKRON
Schrader Bellows Div. of Scovill, Inc. 202 W. Exchange St. .... 50M+

OHIO: CANTON
Bur-Bilt Products Corp. 1221 2nd St. N.E. (Industrial Friction Material, Non-Asbestos Friction Materials, Molding & Bonding) .... NR

OHIO: CINCINNATI
Conway Clutch Co., The 2747 Colerain Ave. (Air Activated) .... 1/4M+

OHIO: CLEVELAND
EATON CORP., AIRFLEX DIV. 9919 Clinton Rd. (ZIP 44144) (Air Actuated) (216-281-2211; For The Name Of Your Nearest Eaton Airflex Representative Call Toll Free 800-243-8160 Ext. 21; In Conn. Call 800-842-0225 Ext. 21) .... 1M+
Electromatic Inc. 3615 Superior Ave., Bldg. 42-T .... 1/4M+

OHIO: DAYTON
Dayton Superior Products Inc. 400-T Linden Ave. (Tooth) 1/4M–
Equipment Concepts M.P.O. Box 2135-T (Tooth) .... NR

OHIO: HAMILTON
Force Control Industries, Inc. 3656 Dixie Hwy. .... NR

OHIO: MENTOR
Great Lakes Power Products, Inc. 7455 Tyler Lane, P.O. Drawer 230 .... 10M+

OHIO: MINSTER
MINSTER MACHINE CO., THE (ZIP 45865) (Combination Pneumatic Friction Clutch & Brake) (419—628-2331) .... 25M+

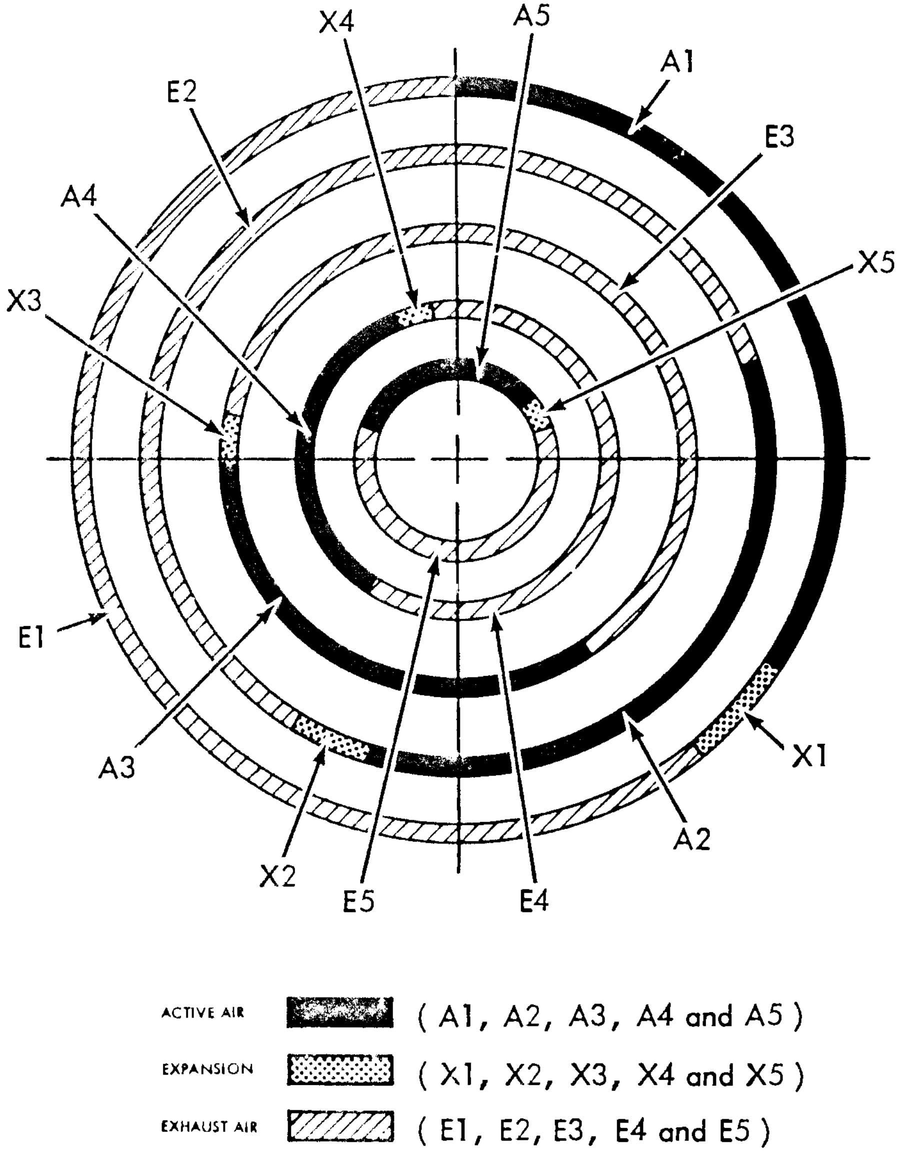
OHIO: TOLEDO
DANA CORP. 4500 Dorr St. (ZIP 43697) (Out Of State Call Toll Free: 800-343-3027) .... 50M+
(See Our Corporate Advertisement In Vol. 10)
♦ *See our catalog in* THOMCAT vols. 12-17

OHIO: WILLOUGHBY (CLEVELAND)
ORTTECH INC. 4399T Hamann Pky. (ZIP 44094) (216—951-5533) .... 50M+

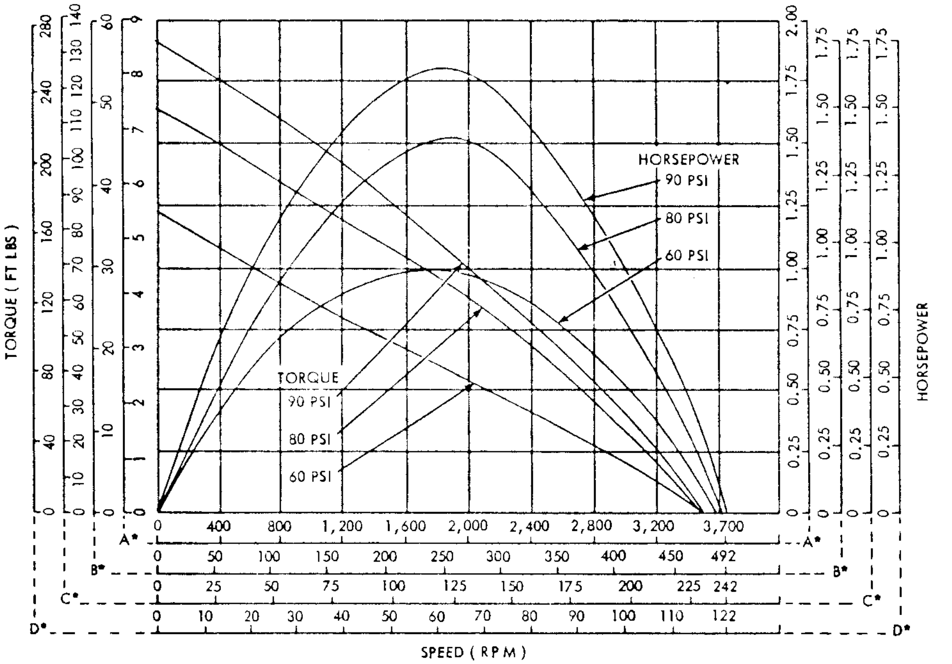
TEX: WICHITA FALLS
Wichita Clutch Co. Industrial Group, Dana Corp. 2800 Fisher Rd. (Air Tube Disc Clutches & Brakes) .... 50M+

WIS: RACINE
Twin Disc, Incorporated 1340 Racine St. .... 5M+

[illegible] showing how the power [illegible] of a 5-cylinder piston motor overlap. The dura- [illegible] is designated as A1, A2, A3, A4 and A5. Expansion is indicated as X1, X2, X3, X4, and X5. Exhaust is indicated by E1, E2, E3, E4, and E5. (Courtesy of Gardner-Denver Co.)



Fluid Power: Pneumatics Olaf A. Johnson, 1978



MOTOR DESIGN	MAXIMUM OUTPUT AT 90 PSI		FREE SPEED RPM	STALL TORQUE FT LBS	STARTING TORQUE FT LBS
	HP	RPM			
A*	1.8	1,850	3,700	8.6	4.0
B*	1.6	246	492	65.0	30.0
C*	1.6	121	242	133.0	61.0
D*	1.6	61	122	262.0	122.0

* To evaluate the four different motor designs A, B, C and D, notice that the horizontal speed lines, the vertical torque lines and the corresponding vertical horsepower lines are shown as A, B, C and D respectively.

Fig. 5–9. Graph showing performance of four different motors of the radial piston type. These motors are designated as A, B, C, and D. Intersections of torque curves with horsepower curves for the four motors indicate speeds and pressures required for maximum torque and horsepower. Notice that torque and horsepower drop to zero at high speeds. (Courtesy of Gardner-Denver CO.)

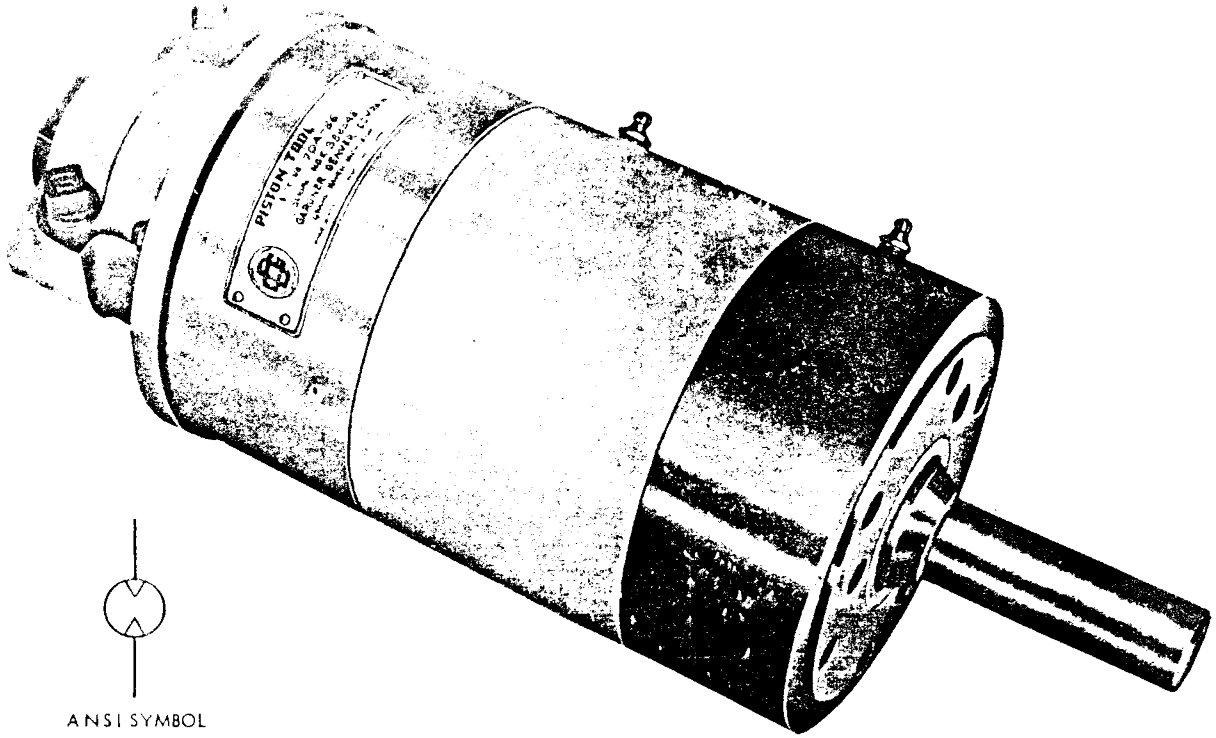


Fig. 5 – 10. Air-operated 5-cylinder axial piston motor. Because these motors are usually designed with five pistons, the overlapping feature of the power impulses is the same as for the radial five-cylinder piston motors as illustrated in the schematic drawing of Fig. 5 – 8. (Courtesy of Gardner-Denver Co.)

Fig. 5–8 is a schematic illustration showing how the power strokes overlap. The duration of active air on each piston is designated by A1, A2, A3, A4, and A5. Expansion is indicated by X1, X2, X3, X4, and X5, and the exhuast is indicated by E1, E2, E3, E4, and E5. Motors of this type have relatively little exhaust noise and, if objectionable, this can be reduced by the [illegible] a muffler.

The chart shown in Fig. 5–9 compares designs of four radial piston motors designated as A, B, C, and D. The torque and horsepower curves show how these characteristics are affected by speed and by various applied pressures. From this chart it is apparent that maximum horsepower in no case is obtained at maximum speed. High torque is an inherent characteristic of piston-type air motors, and because they are mainly used in applications where starting torque and stall torque are important considerations. These are shown at the bottom of the chart, based on maximum output of each of the four motors operated at 90 psi pressure.

Axial Piston Motors. Air motors of the axial piston type, like motors of the radial piston type, are usually designed with five pistons and therefore the overlapping feature of the power impulses are the same as for the radial five-piston motors previously shown in Fig. 5–8. The axial piston motors are available only in small sizes, usually less than 3 HP. See Fig. 5–10.

Like other piston motors they are characterized by fast starting torque, reaching their full operating speed almost instantly. Their mechanical design is quite different from that of the radial piston motors, however. Their small pistons reciprocate axially (parallel to the drive shaft) in sequence. Their piston rods are attached to a wobble plate. The wobble plate, at an angle to the axis of rotation, then causes a rotary motion to a gear train which turns the drive shaft.

Factors that determine the power of these motors are:

1. Inlet line pressure.
2. Number of pistons.
3. Area of pistons.
4. Length of piston stroke.
5. Speed (rpm).

Fig. 5–11 is a cutaway drawing showing the inner construction of the axial piston motor shown in Fig. 5–10. The sectional views in Figs. 5–12, 5–13, and 5–14 show the complete flow of air pressure and exhaust in the sequence for one of the five pistons, which all operate in consecutive order. These sectional drawings are all exploded views, whose sections on both sides of the centerline are not necessarily exactly opposite in the actual motor. The drawings are simplified.

Air Inlet. The air enters the motor at port *A* (see Fig. 5–12) flowing in the direction of arrows through small holes *B* in the distributor bushing *C* into annular groove *D* in the distributor *E*. The groove *D* opens to a slot *F* in the distributor *E*,

thus permitting the air to flow through an elongated slot G in the bushing C and through hole H to cylinder I, putting pressure on piston J.

There are five pistons, so there are five slots G in the bushing C and five holes H, one for each piston. The air pressure then forces the piston J forward. The pistons are connected to the wobble plate K with connecting rods L. There are five connecting rods, one for each piston. The connecting rods are fastened to the piston and the wobble plate with ball bearings M so they

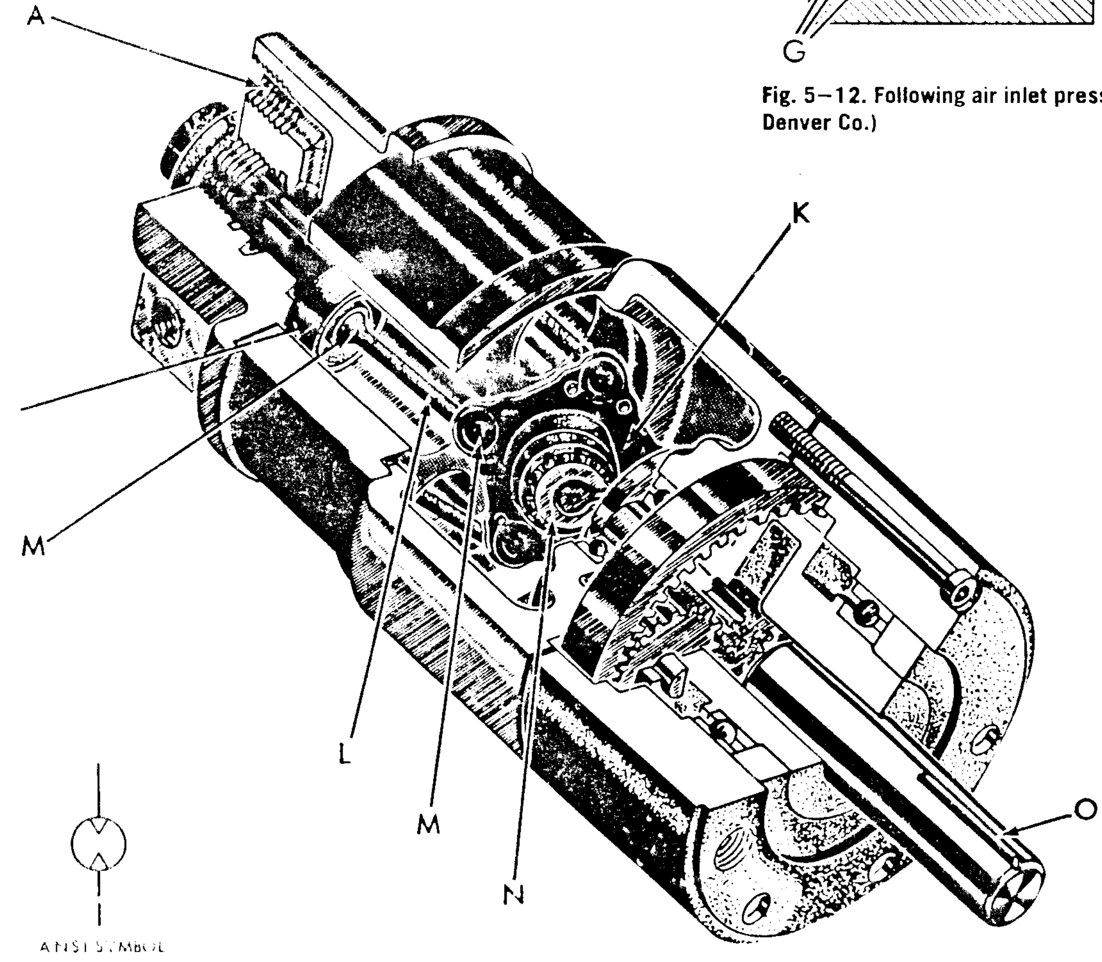


Fig. 5-11. Sectional view of the 5-cylinder axial piston motor shown in Fig. 5-10. (Courtesy of Gardner-Denver Co.)

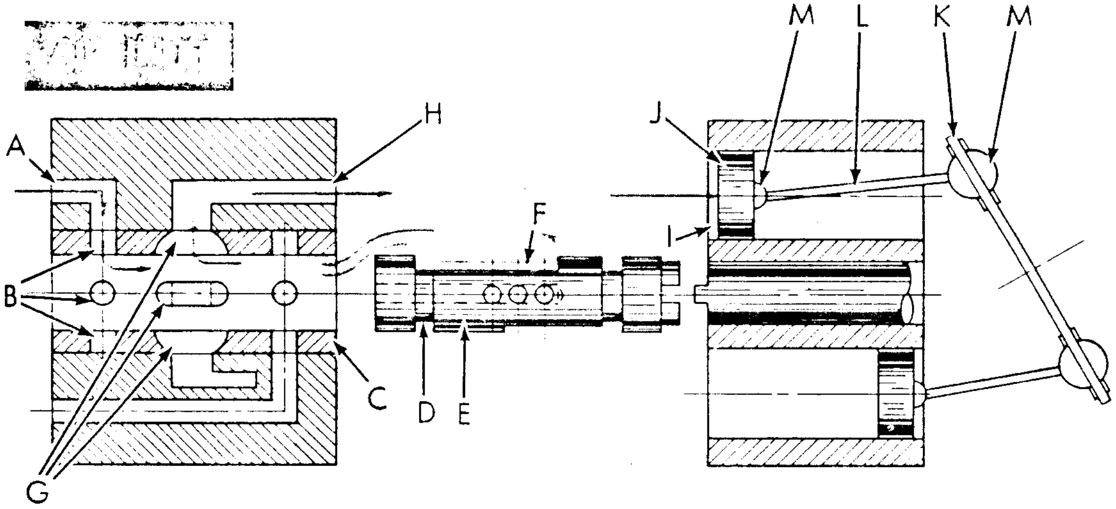


Fig. 5-12. Following air inlet pressure through motors shown in Fig. 5-10 and Fig. 5-11. (Courtesy of Garner-Denver Co.)

may be free to assume a non-axial position. The wobble plate is mounted on ball bearings N on the drive shaft O, as shown in Fig. 5-11. Bevel gears (not visible) are mounted on the cylinder housing and the wobble plate. These gears keep the wobble plate in its proper central position as it imparts rotary motion to the drive and the distributor E. The distributor is connected by a key X and a slot Y, Fig. 5-13.

Primary Exhaust. As shown in Fig. 5-13, when the piston reaches the *nearly-in* position the distributor exhaust holes P will align with a slot Q in the distributor bushing. This permits the air that was used to drive the piston to exhaust to the atmosphere through a hole R in the center of the distributor E. This primary exhaust starts just before the piston J reaches the end of its travel.

Secondary Exhaust. In Fig. 5-14 the piston J is shown at the end of its inward stroke ready to return. Even if all the pressurized air used to drive the piston in had been exhausted at this point,

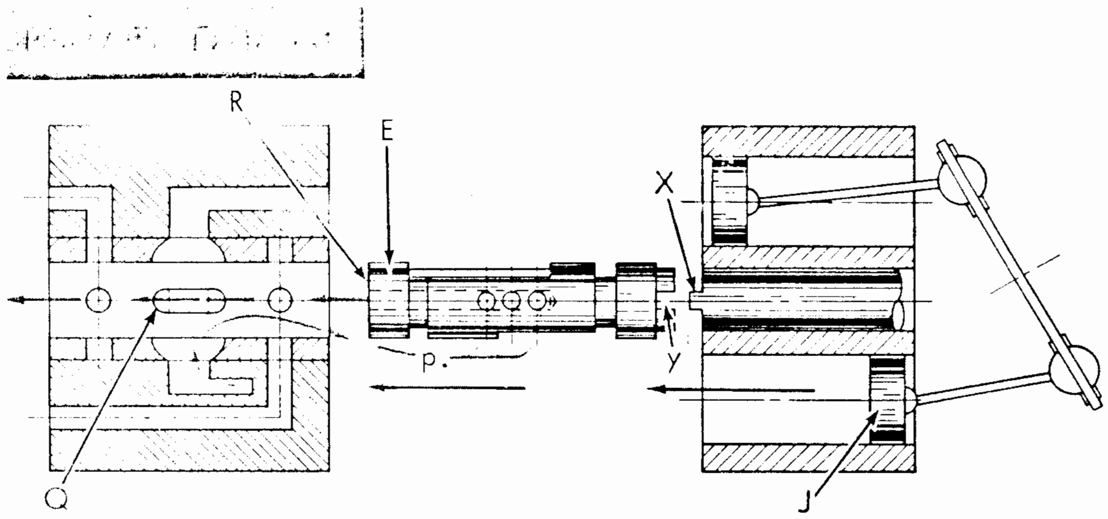


Fig. 5–13. Following primary exhaust of pressurized air used for driving piston in as illustrated in Fig. 5–12. (Courtesy of Gardner-Denver Co.)

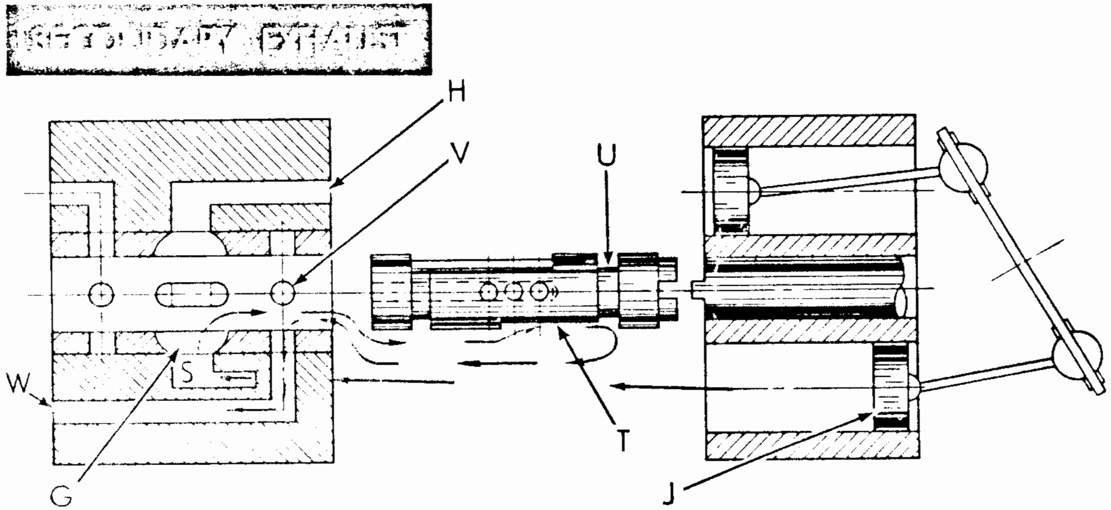


Fig. 5–14. Following secondary exhaust. This is the air subject to recompression if not properly exhausted. (Courtesy of Gardner Denver Co.)

the air present in the cylinder would have a tendency to be compressed as the piston J returns. However, to provide free exhaust to the atmosphere a passageway S is provided (entrance indicated by arrow) in the motor head, where the air is forced out through slot G in the distributor bushing into a slot T in the distributor and into an annular groove U in the distributor.

From this point the exhaust escapes through the small holes V to hole W, which would be the air inlet port for reverse rotation of the motor. Therefore, when the motor is used for one direction of rotation only, the exhaust port must not be plugged because this would result in a great reduction of power caused by considerable back pressure. For reversible motor rotation a 4-way valve will provide the necessary escape for the secondary exhaust air, eliminating back pressure.

The direction of rotation and speed of air motors may be controlled by a variety of valves operated by hand, foot, solenoid, or other electrical or electromechanical means. However operated, the valve should have a full-flow air passage to utilize the full power of the motor. Speed is controlled by the volume and pressure of air admitted into the motor. The speed of any air motor may be regulated from minimum to maximum rpm without damaging the motor. Valves may be operated remotely, either for a single motor or multiple combinations of motors

TRANSACTIONS OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Volume 52, Part 1, page85-89, 1930

Crankless Reciprocating Machines

By ANTHONY G. M. MICHELL,[1] NEW YORK, N. Y.

The general features and properties of the crankless type of reciprocating machine, as compared with the crank-connecting-rod type, are described and illustrated. A few varieties of the crankless type are briefly described, including the author's slant-slipper form of the mechanism. The limitations of the crank type are pointed out and it is suggested that mechanical science is due to supply a type of machine for various present and probable future applications not efficiently served by either purely rotary machines or reciprocating machines of the crank type. A case is presented for the crankless type as satisfying this need.

WHILE the name "Crankless Reciprocating Machines" is not a quite accurate title for the subject matter of this paper, it has to be used in the absence of a better. It is intended to cover both prime movers and driven machines in which the reciprocating motion of pistons is interconverted with the rotary motion of a shaft without the use of cranks. But according to common and convenient usage the term is restricted to machines in which the motion of the pistons is parallel to the axis of the shaft.

The Crank and Crankless Types of Engines

The most fundamental distinction between a crankless engine, so defined, and a crank engine of the familiar type, is that the crank engine has a two-dimensional mechanism, while the crankless engine has a three-dimensional mechanism. That is to say, the essential parts of the crank engine move only in one or more planes at right angles to the axis of the engine shaft. In the crankless engine, on the contrary, motions of essential parts take place in all three of the coordinate directions of space. Another broad and significant difference in the two types of mechanism is seen when we compare multicylinder machines of each class. The typical crankless engine is an essentially symmetrical construction. Each cylinder, or piston, is geometrically indistinguishable from its fellows as regards its relation to them and to the mechanism. For this reason, crankless engines are sometimes referred to as "round engines" or "barrel engines." The crank-connecting-rod engine on the contrary is essentially a linear, or serial, arrangement, and has at most a bilateral symmetry. These differences are immediately apparent on comparison of Figs. 1 and 2, which show respectively the essential moving parts of a typical multicylinder crank-connecting-rod engine and of a crankless engine with the same number of pistons, namely, eight in each case.

We can, and often do, speak in the case of the crank-connecting-rod engine (Fig. 1) of the first, second, or third piston, or the nth piston from the front or rear end. The motion of any such piston is independent of that of each of the others in so far as it may be given any desired phase relation to those of the others by setting the crank corresponding to that particular piston at an appropriate angle.

In the typical crankless engine, on the contrary (Fig. 2), the several pistons are as truly indistinguishable from each other as the several blades of a turbine wheel, and the sequence and phases of the motions of the several pistons are unalterably determined.

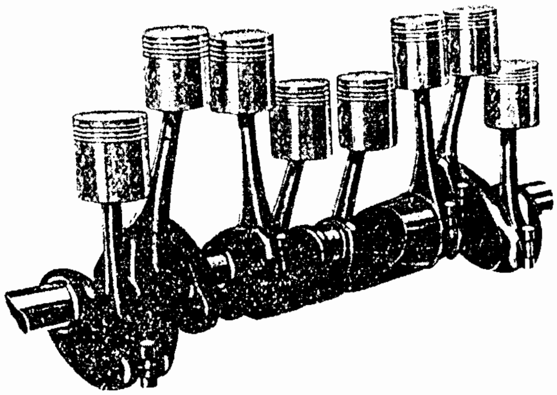


Fig. 1 Essential Moving Parts of a Typical Multicylinder Crank-Connecting-Rod Engine

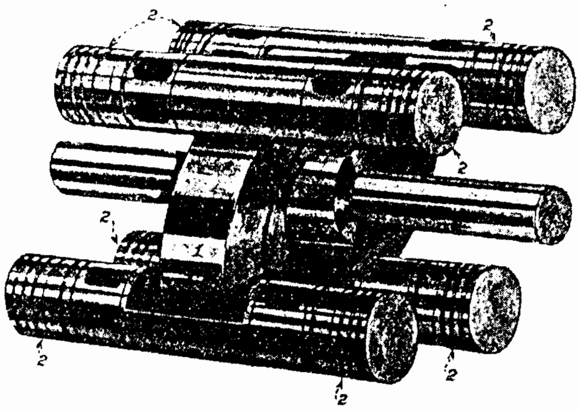


Fig. 2 Essential Moving Parts of a Crankless Engine Having the Same Number of Pistons as the Engine Shown in Fig. 1

Figs. 3, 4, and 5 show respectively three crankless machines, and illustrate three typical forms of crankless construction.

Numerous as are the forms of the crankless mechanism which have been proposed, designed, and patented, many of them extremely ingenious and interesting, these three illustrations cover, so far as the author knows, the only three species of crankless engines which are, or have been, in successful use.

The Ball-Race Type of Crankless Mechanism

Fig. 3 was the first design of the three to be perfected. It represents the well-known "Janney," "Williams," or "Waterbury" transmission mechanism.

[1] Michell Crankless Engines Corporation. Mr. Michell is an Australian engineer, whose academic training was commenced at Cambridge and completed at Melbourne. His engineering work has consisted chiefly in designing and superintending water-supply, irrigation, and hydroelectric power undertakings in Australia and in developing new types of machine elements depending on the Reynolds principle of lubrication. He is the author of several papers published in various European journals in which hydrodynamics and the theory of elasticity are applied to engineering problems.

Presented at a meeting of the Metropolitan Section of the A.S.M.E., New York, April 19, 1929.

Substantially the same machine serves either as a pump or as a prime mover. Regarded as a pump, the rotating shaft 12 carries with it the ring 20, and the stationary inclined ball race 21, upon which the ring 20 rotates, compels the ring to travel in an inclined plane relatively to the shaft. Thus the outer ends of the connecting rods 19 are brought alternately in each revolution nearer to and farther away from the cylinder barrel 15, which comprises an annular series of cylinders such as 15a, containing corresponding pistons such as 18, each attached to the upper end of a rod 19. During one half of the rotation around the race 21, the pistons move outward and draw liquid into the cylinders, while during the other half they move inward and discharge the liquid under pressure, suitable valve action being assumed.

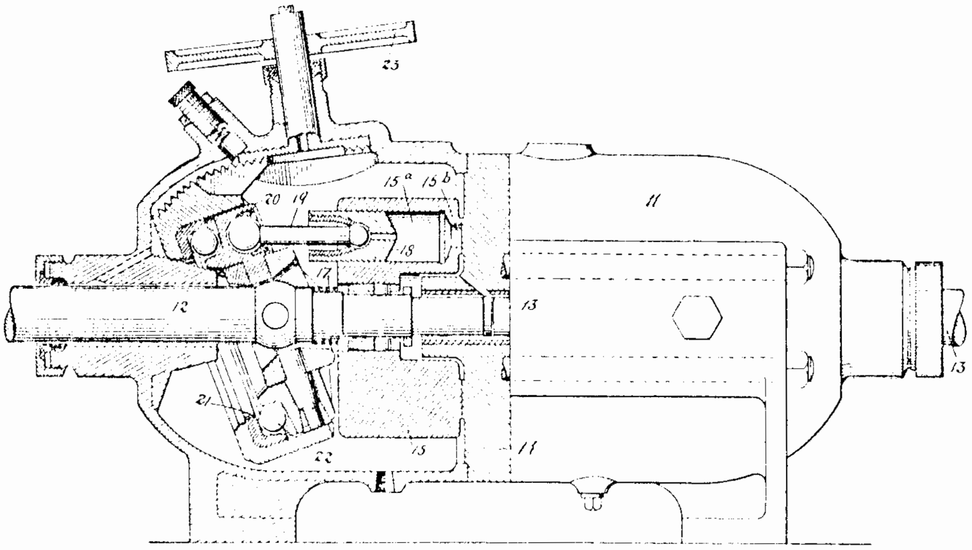


Fig. 3 Ball-Race Type of Crankless Mechanism

An almost exactly similar mechanism, which indeed may form a twin and attached counterpart of the pump, as at 11 in Fig. 3, will function as a motor. The delivery and intakes of the pump and motor being reciprocally interconnected, the first machine drives the second through the medium of the fluid, so that a rotation of the pump shaft 12 causes a rotation of the motor shaft 13. If the pump and motor cylinders are of different sizes, or if the inclinations of the races 21 are different in the two machines, the shafts may have any desired speed relation. Moreover the inclination of the race 21 may be changed, even during operation, with an instantaneous corresponding change of the speed ratio, so that the combined machine serves not merely as a speed-reducing, but as a variable speed-reducing, transmitter.

Before leaving this form of the crankless mechanism it may be noted that the form necessarily adopted for the bearings at the ends of the connecting rods 19, while excellent for exerting a compression in the rod to resist the pressure in the fluid, would be quite unsuitable if there were considerable tension in the rods, such as would arise from inertia effects if the speed were high. The ball race 21, which is necessarily of a considerable diameter, is also only suitable for comparatively low speeds of rotation. This form of mechanism is therefore only suitable for machines operating at relatively low speeds.

The Wabble-Plate Type of Crankless Mechanism

Another practical form of the crankless mechanism is that shown in Fig. 4, the machine illustrated being the "Ali" two-stroke gasoline engine in use as an "outboard" boat engine. Allied forms go under the name of "Arvad engine" and other names. The cylinders 1, which contain the pistons 2 reciprocating in the vertical direction, are in this case stationary while the shaft 3, also vertical, revolves. Each piston is rigidly connected with a scavenge-air piston 4, working in the cylinder 5. Between the power cylinders and the pump cylinders lies the plate 6, which merely "wabbles" or nutates but does not rotate. A ball member 7, fixed to the plate, is socketed in the block 8 which is mounted in the piston unit and is capable of a small extent of transverse motion with respect to it. Thus the nutating motion of the plate corresponds with reciprocating motions of the various pistons in successive phases. Now the plate 6 is mounted by means of a pair of ball bearings, of the combined thrust and journal type, on a flanged collar mounted obliquely on the shaft. Thus as the shaft rotates, the plate, which is prevented from rotating by the ball-and-socket attachments to the pistons, is compelled to execute its nutating motion and so causes the pistons to reciprocate. Conversely, reciprocation of the pistons, compelled to take place in due sequence by their engagement with the plate, causes the latter to nutate and the shaft to revolve. The somewhat unfamiliar type of motion of the nutating plate can be readily followed and calculated when it is observed that its instantaneous motion at every instant is a rotation about an axis which is radially at right angles to the axis of the shaft, and in the same radial plane as the line of greatest inclination of the plate.

The angular velocity of rotation about this axis is, in fact, $\omega \tan \alpha$, where ω is the angular speed of the engine shaft, and α the inclination of the plate.

This nutating motion of the plate with respect to the pistons and frame of the engine may be called a uniform conical nutation, and is of course a three-dimensional motion. Relatively to the shaft, however, it is merely a pure rotation about an axis inclined to the shaft.

It is clear that the point of intersection of the axis of each piston with the central plane of the nutating plate reciprocates with a pure sinusoidal or harmonic motion. The piston itself, however, does not reciprocate in unison with this point of intersection, but with the axial coordinate of the center of the ball 7, which has an arcual and not a linear motion. The motion of the piston therefore deviates somewhat from a true harmonic motion.

Similarly in the Waterbury mechanism, since the radial distance from the shaft axis to the center of the outer ball ends of the connecting rods 19 varies as they traverse their inclined path, the rods 19 take up progressively varying inclinations to the shaft, and the reciprocating motion of the pistons differs somewhat from the true harmonic.

It will be noticed that in both the Ali and the Janney mechanisms there are three relatively moving parts interposed between the shaft and each piston: namely, in the Ali mechanism the ball race, the nutating plate, and the sliding block; and in the Janney mechanism, the ball race, the oblique plate, and the connecting rod. In the Ali engine the ball race is relatively of much smaller diameter than in the Janney gear, and the Ali sliding block, unlike the Janney connecting rod, is equally adapted to carry loads in both directions. The Ali engine is therefore better adapted to run at the high speeds of a multicylinder internal combustion engine than is the Janney mechanism. The load

limitations of the ball bearing in the Ali mechanism, however, seem to limit its use to engines of comparatively small size. Within its limits of application its mechanical efficiency should be very high.

The Slant-Slipper Type of Crankless Engine

The third of the three types of crankless engines which have been mentioned as having reached the stage of actual application is that which the author's own associates have developed. Fig. 2, as already shown, illustrates one example of it. An outline view of the mechanism is shown in Fig. 5. From this

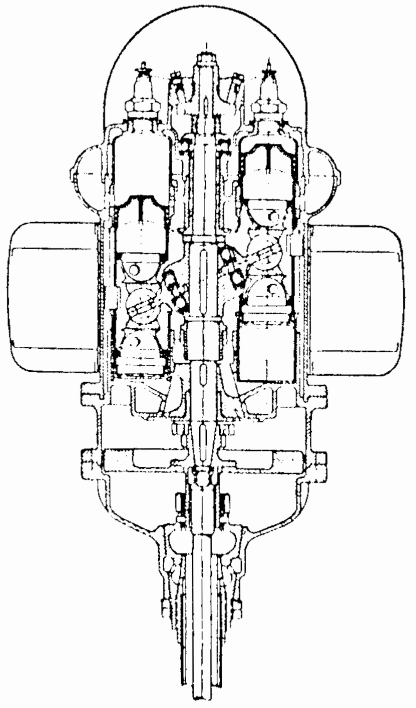


Fig. 4 "Wabble"-Plate Type of Crankless Mechanism

view it will be seen that the shaft has rigidly and permanently mounted upon it an oblique, plane-sided plate 1, which is called the "slant." It has been given this distinctive name since it is not a nutating plate, or "wabble" plate, like the oblique plates in the other two types. It has no other motion than a rotation in common with the shaft. From the slant, motion is transmitted to each of the pistons 2 by a slipper element 3, which serves the purposes of all three of the intermediate elements in the Ali, or in the Janney, mechanism.

Three views of a typical slipper are shown in Fig. 6. It has two bearing faces: a spherical face on one side and a flat face on the other. The spherical face fits into a socket formed in the piston. The flat face makes sliding contact with the flat face of the slant. By means of its spherical support in the piston, the slipper is enabled to follow the continually varying aspects of the inclined face of the slant as the latter revolves; the plane face of the slipper effecting, in fact, a uniform nutating motion of exactly the same kind as the motion of the nutating plate of the Ali engine. During its motion the center of the ball joint remains at a fixed point in the axis of the piston, and at a constant axial distance from the plane surface of the slant. The reciprocating motion of the piston is consequently truly sinusoidal or harmonic.

When in a multicylinder engine several such piston elements are arranged at equal angular intervals around the slant, there is a constant difference of phase between the motion of each piston and that of each of its neighbors.

Usually the piston elements are double, as seen in Fig. 7, pistons on each side of the slant having a common axis and being rigidly connected together by means of a bridge piece 4. A slipper 3 is socketed in each piston so as to effect both strokes of the complete reciprocating unit.

The success of such a mechanism evidently depends on the possibility of the relative sliding motion of the slant and slippers taking place with a minimum amount of friction. Particularly, if the pistons are to drive the shaft, the mechanism being employed as a motor, one recognizes instinctively that the machine would fail to run, or run very inefficiently, if the coefficient of friction of the slipper and plate had ordinary, every-day values.

The solution of the problem lies in the application of the Reynolds theory of lubrication, and the ball-socketed form of slipper which already has been described effects the application. As shown in Fig. 6, the ball is located unsymmetrically with respect to the flat face, to the extent of about one-tenth of the length of the slipper in the direction of the relative sliding motion of the slant. The effective point of application of the load on the slipper is of course the center of the ball. The slipper is thus enabled to function in the same way as the pads of the pivoted type of thrust bearings, which are well known in this country as Kingsbury bearings and which bear the name of the author in Europe. The coacting surfaces require of course an ample supply of lubricant of a viscosity suitable for the

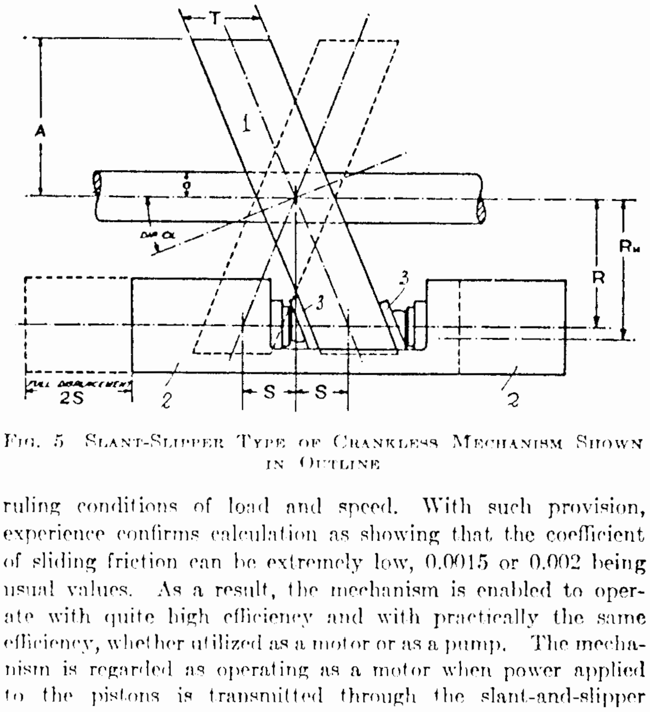


Fig. 5 Slant-Slipper Type of Crankless Mechanism Shown in Outline

ruling conditions of load and speed. With such provision, experience confirms calculation as showing that the coefficient of sliding friction can be extremely low, 0.0015 or 0.002 being usual values. As a result, the mechanism is enabled to operate with quite high efficiency and with practically the same efficiency, whether utilized as a motor or as a pump. The mechanism is regarded as operating as a motor when power applied to the pistons is transmitted through the slant-and-slipper mechanism to rotate the shaft, and as operating as a pump when power applied to rotate the shaft is transmitted through it to cause reciprocating motion of the pistons. Whichever is the direction of power transmission, some energy is of course lost in the frictional resistances of the mechanism; but if the angle of inclination of the slant to the shaft is not too nearly a right angle, such loss of energy in the crankless mechanism is small, and nearly the same in both cases. The crankless mechanism thus conforms to the rule that an efficient mechanism is reversible or, rather, is interconvertible, as motor or "movend;" and conversely, the practical demonstration that one of these machines is in this sense readily reversible is proof that it is mechanically efficient.

To illustrate by symbols: If μ is the coefficient of friction, and θ the inclination of an element of the slant over which sliding is

momentarily taking place, the efficiency operating as a motor is readily shown to be $E_M = 1 - \mu (\tan \theta + \operatorname{ctn} \theta)$; while the efficiency as a pump is

$$E_p = \frac{1}{1 + \mu (\tan \theta + \operatorname{ctn} \theta)}$$

Expanding the latter expression as

$$E_p = 1 - \mu (\tan \theta + \operatorname{ctn} \theta) + \mu^2 (\tan \theta + \operatorname{ctn} \theta)^2 - \mu^3 (\tan \theta + \operatorname{ctn} \theta)^3$$

we see that the efficiency as a motor is less than that as a pump by no more than the fraction

$$[\mu(\tan \theta + \operatorname{ctn} \theta)]^2$$

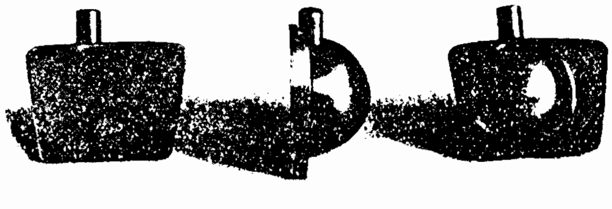


Fig. 6 Three Views of a Typical Slipper

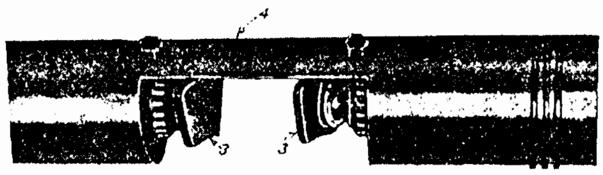


Fig. 7 Double Piston Element of Slant-Slipper Mechanism

If, for instance, $\tan \theta$ has the quite usual average value $^1/_4$,

$$\tan \theta + \operatorname{ctn} \theta = 4.25$$

$$(\tan \theta + \operatorname{ctn} \theta)^2 = 18$$

and if $\mu = 0.002$,

$$[\mu(\tan \theta + \operatorname{ctn} \theta)]^2 = (0.0085)^2 = 0.000072$$

or less than $^1/_{100}$ of 1 per cent, as the difference of the efficiencies in favor of the pump.

Need of the Crankless Type of Machine

With such high efficiency, which has been amply realized in practice, and with the simplicity and compactness of its construction, it is to be supposed that the crankless type may be usefully applicable to a wide range of purposes. Clearly its merits will be most apparent in the case of multicylinder machines, which internal-combustion engines tend more and more to become. What actual advantages it possesses over the types of engines which at present practically hold the field, and in what applications those advantages may be most likely to secure its adoption, the author proposes to set forth by reviewing some special features of the crank engine.

Defects of the Crank Type of Engine

Our familiarity with the crank type, the author believes, is apt to make us oblivious of the real nature of some of its characteristics, and perhaps unduly tolerant of some of its limitations.

In the first place, being essentially a two-dimension machine, the crank engine does not make efficient use of our space of three dimensions. Being neither symmetrical nor compact, it is apt to be unduly extensive, at least in some of its dimensions.

Furthermore, the inherent characteristics of the typical crank engine, namely, that it has no circumferential symmetry and that the motion of its pistons is not harmonic, render it impossible to make it free from vibration by giving it complete dynamic balance; and balance is surely an elementary virtue which the modern engineer, brought up on turbines and electric machines and demanding ever higher rotative speeds, very properly expects every rightly constituted machine to have.

Proceeding to the concrete, and to questions of construction, let us take the essential and characteristic organ of the crank engine, namely, the crankshaft. In an 8-cylinder crankshaft, as shown in Fig. 1, we have a member which is bent at right angles in 32 places. It has 8 crankpin bearings and, in this instance, 5 main bearings. Quite commonly it would have 9 main bearings, or 17 separate bearings in all. Such a shaft in a typical case requires about 160 distinct dimensions to describe it, all of which have to be adhered to with accuracy in all the operations of manufacture. If each of the journals is 3 in. in diameter and 3 in. wide, there will be about 500 sq. in. of bearing surfaces to be machined on these 17 journals, and an equal area of surface to be machined and fitted on the 17 corresponding bearings. This is more than three times the area that has to be machined on the parts performing the corresponding functions in the Michell crankless engine; namely, the slant's two faces, two journals, a thrust collar, and the corresponding bearings and slippers, 18 surfaces in all as compared with 34.

Regarded elastically, the elongated 8-cylinder crankshaft is obviously incapable without the support of its intermediate bearing of supporting the inertia forces of the reciprocating and revolving parts which load it transversely. These forces have therefore to be balanced individually as far as possible (which is to the extent of about 50 per cent), by adding adventitious masses to each crank; and the forces remaining unbalanced in each unit are transmitted through the bearings to the engine frame, to equilibrate as nearly as possible the resultants of corresponding stresses arising from the other units. In so doing, heavy stresses and vibrations are necessarily set up throughout the unit.

While it is true, therefore, that a 6- or 8-cylinder crank engine can be very approximately balanced as a whole, it is only at the cost of much added weight in balance weights, bearings, and frame, and complications to the design, which are entirely unnecessary in the typical crankless engine. What has been said as to the multicylinder crankshaft might be repeated, *mutatis mutandis*, with regard to the connecting rods, to the accommodation of which, by the way, the crank engine usually owes about one-third of its bulk.

Let us proceed, however, to a more vital element, the piston. It is for the sake of the piston, of course, and its unique capacity, with its sealing rings, for compressing in a cylinder a volume of fluid under pressure, that we have reciprocating engines. Turbine rotors will utilize the energy of expanding or falling fluid, with all desirable efficiency and with admirable convenience; but no efficient rotary or other substitute has been found, or seems likely to be found, for the cylindrical piston and its sealing rings, as a compressing agent. Efficient compression is an essential condition of successful operation of the internal-combustion engine, and it is the failure of the gas turbine in this respect which preserves the internal-combustion field for the reciprocating engine. One of the worst of the defects of the crank type is that it does not deal fairly with, or develop the full potentialities of, the piston, especially the piston of the internal-combustion engine.

The secret of the success of the cylindrical piston is, of course, its cylindricality, or the complete symmetry of the piston and the cylinder around their common axis; but in its two-dimensional motion transverse to the crankshaft, the connecting rods apply forces in one diametral plane only of the piston, and this to large extent destroys its symmetry. Contact of piston and cylinder takes place only along the two generating lines in the plane of the connecting rod, from which follows wear of piston and cylinder; while on other portions of the circumference the relatively considerable interspace remains filled with oil or gas, which effectually prevents the conduction of heat from piston to cylinder wall which is an essential function in the internal-combustion engine. An overheated and an unsymmetrically heated and consequently deformed piston is the result, unless the troublesome expedient—almost impracticable except in large engines—of fluid circulation for cooling the piston is resorted to.

In the typical crankless engine, on the contrary, as shown in Fig. 2, since the obliquity of the slant is turned in succession to all azimuths around the axis, every generating line of the piston makes contact at some phase of the cycle with the corresponding generating line of the cylinder. Fig. 8 shows the sequence of the directions of these contact pressures in a particular 4-stroke, 8-cylinder crankless engine, the radial distances to the curves at each azimuth being proportional to the radial pressure between piston and cylinder. This action makes the lubricant in the cylinders effective for reduction of friction, and it also effects symmetrical cooling of the whole piston wall.

In case it may be suggested that the mere "line contact," as it may be called, would not be effective for transmission of the heat, it may be pointed out that in the case of a piston having $^5/_{1000}$ in. diametral clearance in its cylinder, the clearance space is less than $^1/_{10,000}$ in. wide throughout an arc of more than 30 deg., and that the resistance of a film of oil of that thickness is no greater than that of $^1/_8$ in. of metal.

Thus while it is generally agreed that in the conventional internal-combustion engine only the gas rings of the piston are effective for conducting heat from the piston to the cylinder walls, in the crankless engine the whole piston barrel is effective for the purpose. In thus fully utilizing the virtues of the piston, the crankless type admits of higher compression pressures and temperatures, and higher piston speeds, just as its perfect balance admits of higher rotative speeds than the crank type.

Fields of Application of the Crankless Type of Engine

It is believed, therefore, that we can confidently expect a future, especially in the field of the internal-combustion engine, for the crankless type in whichever of its present or future forms may prove to be the best. Its success will come first where the defect and limitations of the crank type hinder its further progress. And the author ventures to suggest that there are clear indications, in several directions, of important new fields which the crank type cannot satisfactorily fill.

Diesel Locomotive. First, there is the oil-engine locomotive for main-line service. Very many attempts have been made with the crank type for the past 15 or 20 years. In the most recent examples 1200 to 1500 hp. has been reached, but the crank engine of this power attains the limit of height of most railway loading gages, and the weight and cost of the structure prevent economic competition with the steam locomotive.

The author visualizes the oil-engine locomotive of the future, of any required power, with a crankless engine of approximately the same size and shape and in much the same location on the locomotive as the present steam boiler, coupled to an electric generator in roughly the same location as the present firebox. The higher speeds which are acceptable for the crankless type will enable the weight and cost to be kept within reasonable limits.

Diesel Aero-Engine. Another field of even greater importance which he sees marked off for the crankless engine of the future is that of the heavy-oil engine for flying machines. The disabilities of the crank type of engine for this application are its inadaptability to the two-stroke cycle—at least without a second crankshaft—its excessive height or other dimension transverse to the shaft, its relatively great frontal area in all its lighter forms, and its deterrent weight. It is true that remarkable progress has lately been made in the reduction of weight of crank engines for this purpose, but whatever success is achieved in that direction will enable crankless engines to be constructed by the same methods at least 30 per cent lighter, and in this field the demand for the lightest possible engine will always be insistent.

Twelve-Cylinder Automobile Engine. Looking in another less lofty direction, if the fashion in automobilism should trend to a

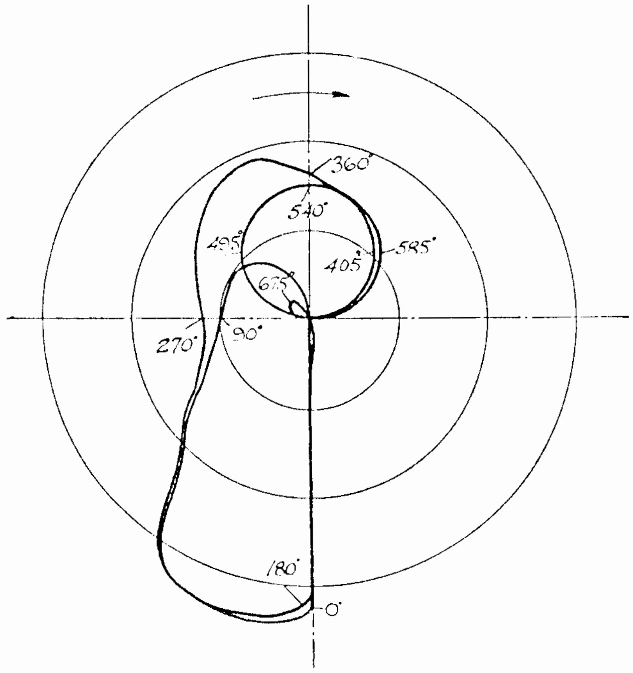


Fig. 8 Showing Sequence of Directions of Contact Pressures in a Particular 4-Stroke 8-Cylinder Crankless Engine

demand for an engine with twelve or more cylinders, as seems very probable in view of the diminution of torque fluctuation—which is now the most serious cause of vibration and unsteady running in automobiles of the best class—the crankless type must apparently be adopted. A 12-crank shaft can scarcely be contemplated with equanimity, and no other crank type than the straight-line type seems to be acceptable. A 12- or even a 16-cylinder crankless engine presents no difficulties, and would be no longer than an 8-cylinder engine.

Compressors for Large Volumes. There are, the author thinks, numerous other fields in which the progress of internal-combustion engineering depends upon the adoption of an engine mechanism different from the crank and connecting-rod, but of those he will mention only one more. It is the field of compressors for very large volumes of air and gas required to be delivered at moderate or high pressures; that is, pressures from 5 lb. per sq. in. upward and volumes of several or many thousands of cubic feet per minute. Considerations of space occupied and the convenience of direct-electric driving demand that the machine shall have a high rotational speed.

The turbine and centrifugal types are negatived not only by their poor efficiency at any but low pressures, but by their inflexible control characteristics.

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THE AXIAL ENGINE

A Promising Type of Exceptional Compactness. Survey of Recent Designs

By Colin Campbell, M.Sc., A.M.I.A.E.

AS a layout, the axial arrangement of cylinders has a number of obvious advantages to recommend it. There are, however, correspondingly difficult problems to solve. Chief of these centres in the anchorage of the wobble plate, because of its complex non-planar motion.

This motion was not understood, complete ignorance virtually prevailing as to the true kinematics. Only recently has this matter been correctly appreciated. This new knowledge should greatly influence developments in engines of this type.

THE compact layout of the axial engine has attracted countless inventors and designers since D. K. West patented the first "swash plate" steam engine (Fig. 1) in 1875. Fortunes have been sunk in developing this mechanism since the advent of the automobile and the aeroplane created a demand for streamlined compact engines. The way of the "wobble plate" engine designer in particular has been incredibly hard. Only now that the kinematics of the problem are clear is it obvious why early designers, with their ignorance of the complex motion of the wobble plate, suffered so many defeats. The first wobble plate engine to take the air revealed a complete ignorance of wobble

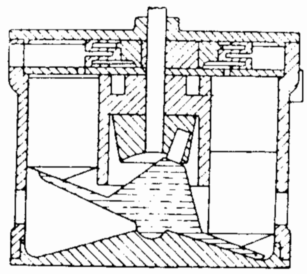


Fig. 1. West steam engine, patented in 1875.

plate kinematics and resulted in seven broken connecting rods immediately after the flight. This engine was a 7-cylinder Redrup "Fury." and when installed in a Simmonds Spartan a successful flight was performed by Captain M. L. Bramson in 1930.

As the name suggests, the axial engine is essentially an engine with the cylinders arranged parallel to, and radially around, the crankshaft. It is possible to have both radial and rotary arrangements, but all modern axial engines are confined to the radial type. To the designer, however, the chief difference lies in the two methods of converting the reciprocating motion of the pistons into the rotary motion of the drive shaft.

The swash plate engine

The first type, the swash plate engine, has a disc integral with the drive shaft and set at an angle so that the axially

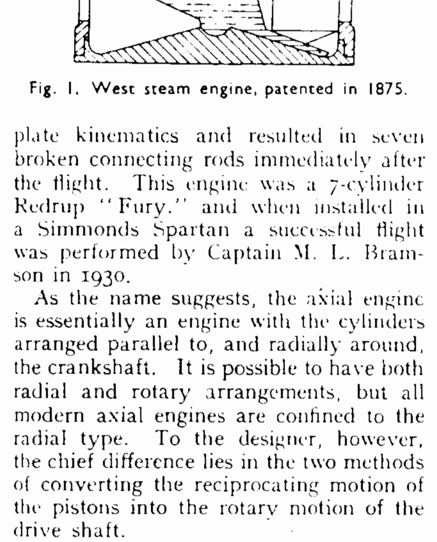


Fig. 3. Diagram of swash plate engine.

arranged piston rods may operate on it to produce the rotary motion. A design that holds much promise is the Michell engine, the swash plate of which is illustrated in Fig. 2. One of the first versions of this engine was designed by Mr. T. L. Sherman, of the Michell Crankless Engines Corporation, and built at Canton, Massachusetts. This was a two-stroke opposed-piston diesel engine, incorporating the form of thrust pad invented by A. G. M. Michell, of Melbourne, Australia. It gave a b.m.e.p. of 95 lb. per sq. in. at 900 r.p.m. The Canton engine is the basis of the marine diesel engine made under licence by the Sterling Engine Company, of Buffalo. This firm is now in production on two engines, both of which have four cylinders and eight pistons working on the opposed-piston principle.

The smaller engine has a bore and stroke of 3½in. by 4¾in., and will give a continuous rating of 75 b.h.p. at 1,200 r.p.m. The larger one delivers a rated horse-power of 135-150 at 1,200 r.p.m. from a bore and stroke of 4½in. by 5in. In the near future a 375 b.h.p. engine of 6½in. bore and 8½in. stroke will be available. The Sterling Engine Company have now more than 30,000 hours of test-bed experience, and claim to have reached a high standard of reliability. The construction of these Sterling engines follows the basic Michell principle, having independent reciprocating units contacting a plane-faced slant or inclined disc. Co-action with the slant is through Michell type slipper pads.

The reciprocating units are made up of guided bridge members, which embrace the slant, with short piston rods connecting pistons and rods. Fig. 3 shows the arrangement of the bridge members and slipper pads. The Michell or Kingsbury bearing used in this type of engine consists of a thrust collar integral with the shaft running between two rings of pivoted segmental pads. Each pad is supported on a pivot situated slightly behind the centre of pressure. The position of the pivot causes the pad to tilt and ride on a self-generated wedge of oil in much the same way as a surfboard on an Atlantic roller. In the swash plate engine the thrust collar becomes the slant, and the slipper pad pivots are spherical cups carried by the bridge members. A tail attached to each slipper pad serves to keep it in correct alignment.

A swash plate engine with three or more reciprocating units may be balanced perfectly. It can be shown mathematically that the summation of all piston inertia-moments about an axis perpendicular to the shaft is of constant magnitude and revolves around the shaft at engine speed. In Fig. 3 it is seen that the natural tendency of the slant is to swing to a position normal to the shaft. By careful design it is possible to arrange that these two opposite moments are equal, thus giving perfect balance.

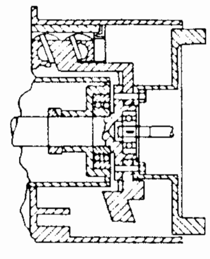


Fig. 2. Michell swash plate engine.

The equation for balance is:—

$$M_s = 2nm_1 \frac{C}{A^2 + a^2}$$

in which

A is the outer radius of the slant;

a is the radius of the symmetrical hub of the slant;

C is a coefficient greater than 1, such that $C.r = r_m$ (r_m being the radius of the centre of mass of the reciprocating unit);

M_s is the mass of the slant;

m_1 is the mass of one reciprocating unit;

n is the number of reciprocating units;

r is the cylinder-circle radius.

As an aero-engine the Michell engine appears to be inherently heavy. In an early 8-cylinder automobile engine of 3in. bore and 3½in. stroke, the slant weighed 40 lb., and the piston weighed 7·2 lb. Until methods of reducing this weight are found the Michell engine will probably find its field of usefulness confined to

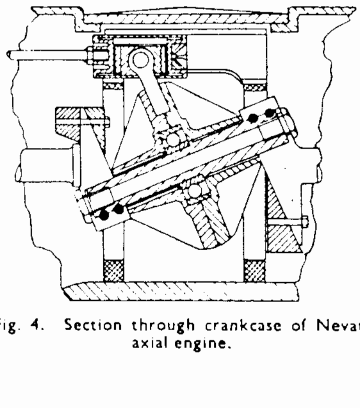


Fig. 4. Section through crankcase of Nevatt axial engine.

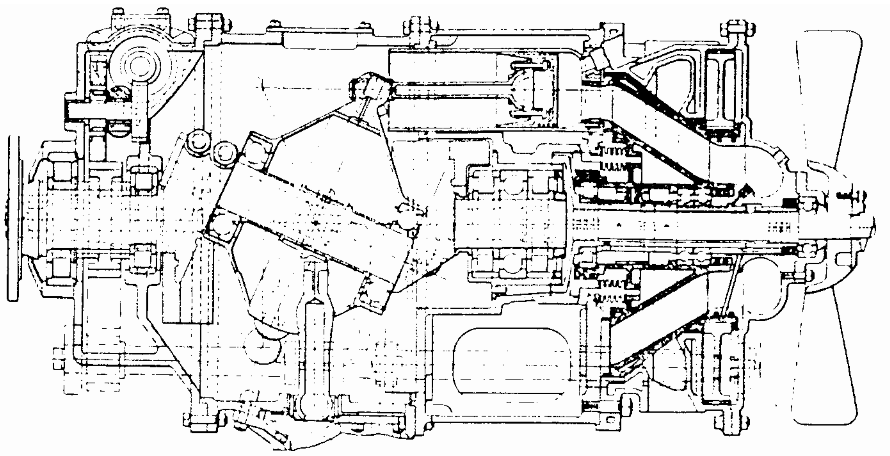


Fig. 5. A nine-cylinder axial engine developed by Bristol Tramways and Carriage Co., Ltd.

stationary installations. This is not, however, the opinion of the Michell Crankless Engine Corporation, as their chief designer has recently completed designs for a two-cycle diesel aircraft engine delivering 2,000 b.h.p. at 2,000 r.p.m., weighing 2,100 lb., and having a diameter of 35in. and a length of 75in.

The wobble plate engine

Owing to lubrication difficulties with the swash plate engine, engineers adopted the expedient of reducing the diameter of the swash plate until it took the form of an inclined crank pin supported by two webs, as shown in Figs. 4 and 5. Running on suitable bearings on this pin was a member constrained in such a way as to wobble backwards and forwards and not have the rotary motion of the crankshaft. The connecting rods were connected by some form of universal joint to the periphery of this wobble member. The peripheral speed of the thrust bearing was thus reduced, since the thrust could now be taken at the crank pin instead of at the radius of cylinder centres.

The wobble plate engine, as this arrangement is called, has its own peculiar problem in the method of anchoring or stabilising the wobble plate. It is hardly necessary to point out that before the torque can be delivered to the crankshaft the wobble plate must be supported in such a way as to transmit the torque reaction to the crank case. A perusal of British Patent Specifications reveals a variety of suggested forms of stabiliser. Most of these show a complete ignorance of the kinematics of wobble plates.

A common mistake is to provide straight guides arranged around the crankcase in the fond hope that suitable extensions round the periphery of the wobble plate will operate between such guides. If any of these designs had materialised it would have been found that it was impossible to rotate the crankshaft without breaking the wobble plate, the straight guides, or the crankcase.

For successful control of the wobble plate, it is vital to have a complete understanding of its natural motion. It is less than eight years since the experimental staff of the Bristol Tramways and Carriage Company, Limited, under the supervision of Major C. G. Nevatt, O.B.E., made the discovery of the figure-of-eight path traced out by all points on the wobble plate. In Fig. 9 a wobble plate with four ball connectors is shown, and it would be natural to assume that the centre of each ball moves in a straight line over the engine centre line. This is the mistake made by many patentees. The path of every ball connector is a form of lemniscate or figure-of-eight. All points on this lemniscate are equidistant from the centre of the wobble plate. Hence the lemniscate is not contained in one plane, but must be considered as if described on the surface of a sphere.

The end elevation of the wobble plate shows that the centre of each ball connector describes an elliptical path, making two complete circuits of this path for every single revolution of the engine. It is a matter of simple observation to discover that any lemniscate drawn on a sphere presents an elliptical end elevation. Thus one lemniscate in particular with a certain ratio of chord to length will present a circle in end elevation. When this was first realised eight years ago it was hoped that the path described in end elevation by points on the periphery of the wobble plate was this particular case. The path of one of the operative arms of the wobble plate on one of the experimental axial engines made by the Bristol Tramways and Carriage Company was measured. It was a perfect circle. A mathematical proof is given in Appendix I.

The early wobble plate engines made by the Redrup Lever Engine Syndicate and the Bristol Tramways and Carriage Co., Ltd., were stabilised by a ball-jointed arm attached to the inside of the wobble plate, as shown in Figs. 5 and 15. On careful examination it is seen that this method makes no provision for the required figure-of-eight motion. The portion of the wobble plate adjacent to, and in the same plane as the stabiliser arm, is constrained to move in an arc of a circle while the portion of the periphery farthest from the arm describes a wide figure-of-eight with a chord twice the natural size. Intermediate points have figure-of-eight paths, the chords of which increase in proportion to their distance from the stabiliser arm. That this was an unsatisfactory form of stabiliser was realised, and after much research by the Bristol company a form of arm was evolved which had its outer extremity located in a slide in the crankcase and was reciprocated at twice engine speed by a connecting link and eccentric. Careful measurement on one of these engines confirmed that the figures-of-eight on all cylinder centres had identical chords and that the end elevations of the paths were perfect circles.

Early designs

The Redrup "Fury" aero-engine, which was first exhibited at the Aero Exhibition in 1929, was a five-cylinder aero-engine of 3in. bore and 5in. stroke. This engine of 60 b.h.p. had a maximum diameter of 18in. and had a weight-to-power ratio of 2·6. It was fitted with cast-iron cylinder barrels shrunk into aluminium heads. Each cylinder head had one exhaust and two inlet valves operated by rocker arms and push rods. The tappets were actuated by two-lobe cams running at one-quarter engine speed and driven in a counter clockwise direction by epicyclic gearing from the crankshaft. The connecting rods had spherical joint attachments to the pistons and very compact Hooke's joints where they were attached to the Star-member, as the Redrup wobble plate was called.

The torque reaction was taken by a torque arm universally attached to the crankcase. A later modification of this design, patented by Captain M. L. Bramson in 1931, had a torque arm attached to the periphery of the wobble plate and spherically mounted in the crankcase. This again gave an imperfect stabilisation, but offered greater simplicity than the original Redrup engine. The wobble plate could be a sturdier structure, since it was no longer split to permit entry of the torque arm.

The Bristol Tramways' engines

Successful stabilisation was not achieved until the Bristol Tramways and Carriage Company realised that the torque arm should be reciprocated at twice engine speed to impart the correct figure-of-eight motion. It was at first thought that it would be sufficient to allow the spherical mounting of the torque arm to move freely in a straight guide in the crankcase arranged transversely below the wobble plate. The torque arm, however, is at a mechanical disadvantage in moving its

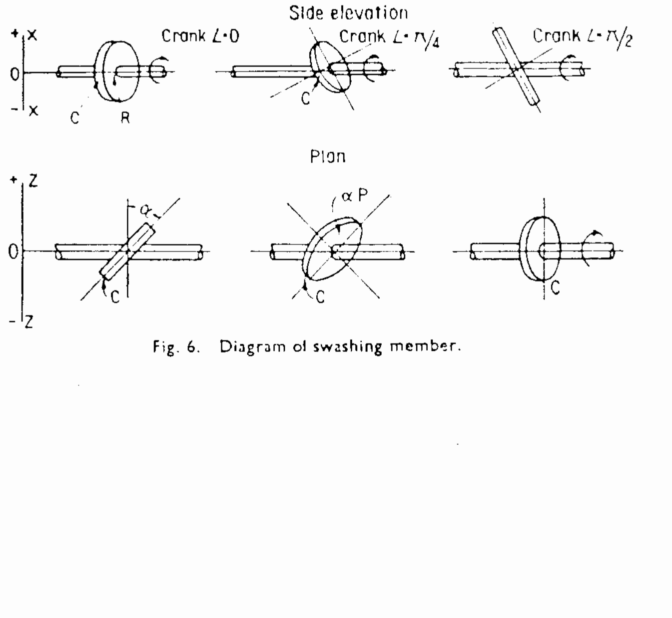


Fig. 6. Diagram of swashing member.

crankcase extremity against the frictional force of the guides and its own inertia. It was found that the arm fractured at its inner extremity due to fatigue in quite a short time. The only remedy therefore was to connect at right angles to the outer extremity a short link which was reciprocated by an eccentric rotating at twice engine speed. This was found to give the required motion to the wobble plate within quite small limits.

The engine with this modified stabiliser, the first of several types, was designed for installation in a bus chassis. The engine had nine cylinders of 7 litres capacity developing a b.h.p. of 148 at 3,000 r.p.m. The standard J.W. type six-cylinder bus engine of similar capacity developed 116 b.h.p. at 2,600 r.p.m. The orthodox valve system was replaced by a low-speed rotary valve driven at one-eighth engine speed. This valve is seen in Figs. 5 and 16, and consists of a single casting in the form of a truncated cone incorporating eight ports and passages and water cooling space around these gas passages. An ingenious arrangement was employed to control the pressure on the gas seal. The cylinder head exhaust and inlet ports were fitted with small venturi tubes. These venturi tubes prevented the cylinder pressure from forcing the sleeves too

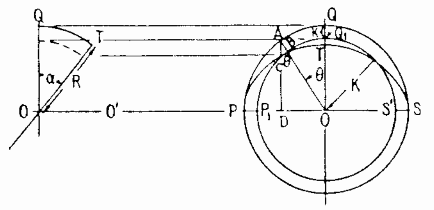


Fig. 8. Operative circle of the swash plate.

tightly against the valve face, since the cylinder pressure also acts inwards on the projected areas of the outer bell-mouthed portions. These sleeves were made of phosphor-bronze and were brazed at the outer ends into spring-loaded flat plates, forming together a complete annular seal ring.

The rotary valve was a steel casting containing water passages surrounding each gas passage. Two "packless" or bellows glands constituted the water passage between the moving valve face and the stationary cylinder face. Although this arrangement is unorthodox practice, and therefore suspect, no water leakage occurred during the extensive tests carried out on the engine. Defective valve castings were, however, one of the troubles experienced during development. With all rotary valves the critic looks for lubrication difficulties, but when a suitable pressure plate had been evolved very little overheating or seizing took place. An engine was even run for fourteen hours without lubrication to the rotary valve. Several thousands of an inch were ground away from the valve face, but the engine did not miss a single power stroke during the test.

A marked advantage of the wobble plate engine when compared with the J.W. type, a six-cylinder seven-bearing engine, was revealed by motoring tests. The axial type required only one-quarter the power when motored over at the same

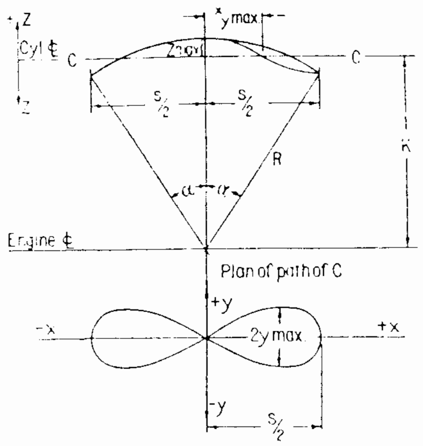


Fig. 7. Mechanical limitations of the wobble plate.

speed as the in-line engine. Although of the same capacity, the J.W. weighed 12 cwt., and the axial engine weighed 7 cwt.

The Nevatt engine

The wobble plate engine, for which the patent rights are held by Nevatt Axial Engines, Ltd., is shown in Fig. 4. This design has a type of stabiliser, invented by Major C. G. Nevatt, which combines the duties of a stabiliser with those of a crosshead. Reference to Figs. 10 and 20-24, which give the details of the stabiliser, show that the figure-of-eight motion is accommodated by ball sockets around the wobble plate which are mounted eccentrically in cylindrical crossheads. The eccentricity is made equal to the radius of the natural circle for the particular wobble plate. There are several ways in which this natural lemniscate can be accommodated, but all these methods can be grouped into three fundamental classifications:—

(*a*) With universal attachment of wobble plate entirely inside the eccentric roller.

(*b*) With eccentric roller entirely inside the universal attachment.

(*c*) With one axis of freedom of universal joint inside the eccentric roller and the other axis outside.

The type shown in Fig. 10 is in the first group, since its universal attachment, the spherical joint, is internal. Provided the engine has three or more cylinders, this arrangement will absorb the torque reaction while allowing the wobble plate to follow its natural motion with the minimum of backlash. The crosshead stabiliser provides a solution to one of the major problems of the wobble plate engine, excessive oil consumption. The swashing motion produces a pronounced centrifugal effect on the lubricant fed to the centre bearing of the wobble plate. This effect is not simply due to the force

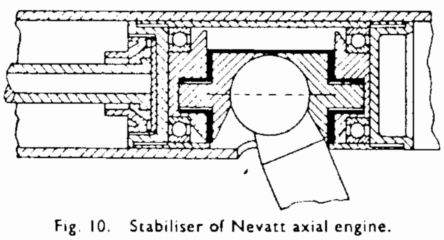


Fig. 10. Stabiliser of Nevatt axial engine.

associated with the constant acceleration of true circular motion, but would seem to be largely due to the inertia force caused by the reversal of motion at each end of the stroke. This centrifugal effect causes the lubricant to be injected in copious quantities into the cylinder with the customary excessive oil consumption associated with this condition. The crosshead stabiliser provides a piston rod motion, i.e., with no angularity, and enables the outer extremities of the cylinders to be isolated from the crankcase by piston-rod glands.

Applications of axial engines

The swash plate engine and the Nevatt wobble plate engine have a field of application as wide as the orthodox crank and crosshead engine. They may be used as double-acting engines or single-acting two-strokes or four-strokes with the underside of the piston used as a compressor. The simple wobble plate engine, i.e., the Redrup, Bramson, or Bristol type, owing to its connecting-rod angularity, cannot utilise the underside of the piston, but it shares with the other two types many attractive features when used as an opposed-piston two-stroke similar to the Junker's "Juno." Both types of wobble plate engine are more adaptable to high-speed

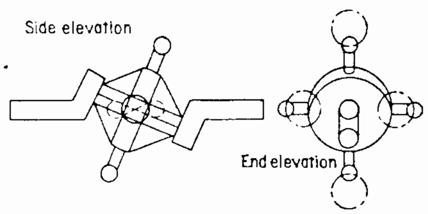


Fig. 9. Wobble plate with four ball connectors.

duty than the swash plate engine owing to their lighter reciprocating parts.

The axial engine offers an ideal layout to the aero-engine designer. A 1,000 b.h.p. aero-engine would not exceed 30in. in diameter. Even an overall diameter of 2ft. seems possible when the experience of experimental designs becomes available. Mr. T. L. Sherman anticipates a maximum power of 2,000 b.h.p. from his swash plate aero-engine of 35in. diameter. Major C. G. Nevatt has recently designed a 2,000 b.h.p. engine with a maximum diameter of 34in. and an overall length of 52in. This engine has a total cylinder capacity of 68 litres, and the front area is only 13·5 sq. in. per litre, or 0·45 sq. in. per b.h.p. A certain modern liquid-cooled aero-engine has a frontal area of approximately 23 sq. ft. per litre, or 0·59 sq. in. per b.h.p.

As an air-cooled engine, the axial arrangement of the cylinders gives excellent air distribution without much recourse to baffling. The maximum allowable cylinder head temperature imposes a limit on radial and in-line engines which is seriously reduced by the rear of the head receiving a lower mass flow than the front. With good design and suitable longitudinal finning the cylinder heads of the axial engine receive the full benefit of the air stream with equal flow to all parts of the head. The more modest cooling demands of the cylinder barrel are supplied by the deflection of the air stream from the head

through longitudinal fins. It must be ad- [illegible] the radial strength given by normal transverse finning, and for equal pressures an air-cooled axial engine cylinder barrel will always weigh more than a modern radial engine cylinder barrel. Weight saving in other directions would, however, more than compensate for this loss.

The vibration stresses induced by the increasing powers required for higher speeds are taxing the brains of installation designers. An engine mounting which is massive enough to absorb the vibrations and residual out-of-balance forces from a modern engine becomes painfully heavy to the eyes of the designer. The relatively small mass of the modern aero-engine offers little assistance in damping out its own considerable power impulses. These difficulties would be alleviated by the excellent balance of the axial engine. All out-of-balance inertia forces can be perfectly balanced by suitably arranged balance weights, and only the power impulses remain to be absorbed by the engine mounting and the airframe. Appendix IV suggests a graphical method for obtaining the sizes of the required balance weights

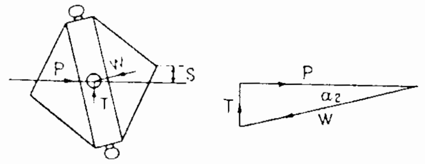


Fig. 11. Diagram of piston thrust.

in the wobble plate engine. The equation for the mass required in the slant of a swash plate engine for perfect balance has already been given.

The future of the engine

If our present knowledge of the natural motion of the wobble plate had been discovered twenty years ago the wobble plate mechanism would have taken its place by the side of the normal crankshaft as a standard method of converting reciprocating into circular motion. In the aviation world

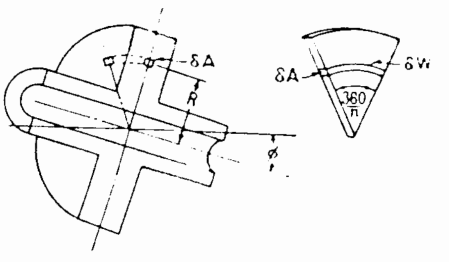


Fig. 12. Section through simple form of wobble plate.

it might easily have completely ousted the normal crankshaft by virtue of its compactness. It may still not be too late, but the axial engine is now in the position of a sprint runner who has made a bad start. The vast amount of combined knowledge from research and development incorporated in a modern aero engine is almost inconceivable. Much of this knowledge in the form of pure research would be immediately available to the wobble plate designer, but years of concentrated development work would be necessary before a design could be produced with a chance of survival against such masterpieces as the Rolls-Royce "Merlin" and the Bristol "Hercules."

APPENDIX I

A wobble plate engine is a specific type of swash plate engine "wherein the engine shaft is provided with an inclined crank pin which is rotatable in the swash member and wherein means are provided for so constraining the movement of the swash member that all points in a plane at right angles to the inclined crank pin and at equal distances therefrom traverse different paths of the same shape and wherein the connecting rods of the engine are articulately connected to the swash member."

That these paths are all figure eights when viewed in plan and present circular paths when viewed in end elevation will now be proven and the equation of the two views evolved.

Partial proof

The mechanical limitations of the wobble plate constitute the first condition. The distance of any point on the periphery of the wobble plate from its centre is constant. The path is thus always on the surface of a sphere of radius R. (See Fig. 7.)

In terms of co-ordinates z and y at right angles to the engine shaft, the equation of the [illegible]

$$y^2 + z^2 = K^2 \qquad r = K^2$$

where $K = R/2\,(1 + \cos\alpha)$

and $r = R/2\,(1 - \cos\alpha)$ = radius of natural circle.

For simplicity the wobble member (non-rotating) is considered as a thin cylindrical shell surrounding the periphery of the swashing member (rotating) which, by virtue of this fact, is constrained to the plane of the swashing member in all crank positions.

In Fig. 6 the swashing member is shown viewed in plan and side elevation in angular positions of o, $\pi/4$ and $\pi/2$ radians after T.D.C. Consider that with each cylinder centre there are points C_1, C_2, C_3, etc., which are situated on the wobble member and which during each cycle follow separate but similar paths around their respective cylinder centres. Each point has, however, a phase difference from its neighbouring points.

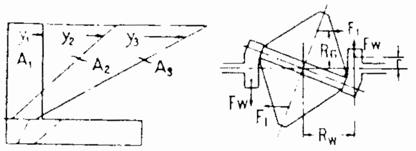
R = actual radius of wobble plate.

R_p = projection on R on vertical plane through axis.

α_p = projection of α in horizontal plane through axis.

As crank angle nt changes from o to $\pi/2$, α_p changes from α to o.

$$R_p = R \sin \alpha_p$$
$$y = R_p \sin \alpha_p$$
$$= R \sin^2 \alpha_p$$



Figs. 13 and 14. Diagrams showing inertia forces.

It is observed that $\alpha_p = \alpha/2$ at y_{max}

$$= R \sin^2 \alpha/2$$

$$y \text{ max.} = R\,(1 - \cos^2 \alpha/2)$$
$$= R/2\,(2 - 2\cos^2 \alpha/2)$$
$$= R/2\,\{1 + (\sin^2 \alpha/2 + \cos^2 \alpha/2) - 2\cos^2 \alpha/2\}$$
$$= R/2\,\{1 - (\cos^2 \alpha/2 - \sin^2 \alpha/2)\}$$
$$y \text{ max.} = R/2\,(1 - \cos\alpha)$$
$$= z \text{ max. (see Fig. 7).}$$

This shows that the paths of the points C_1, C_2, C_3, etc., pass through four points, equidistant by $R/2(1 - \cos\alpha)$ from their respective cylinder centres, twice per revolution.

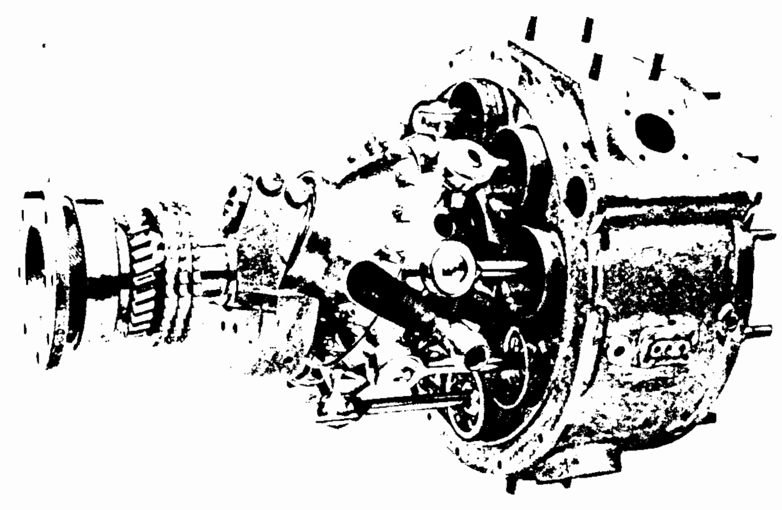


Fig. 15. Wobble plate assembly, Type R.R.3 Bristol engine.

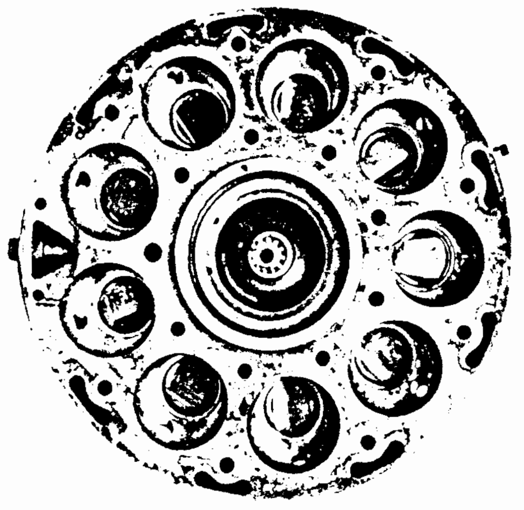


Fig. 16. Cylinder-head, Type R.R.2.

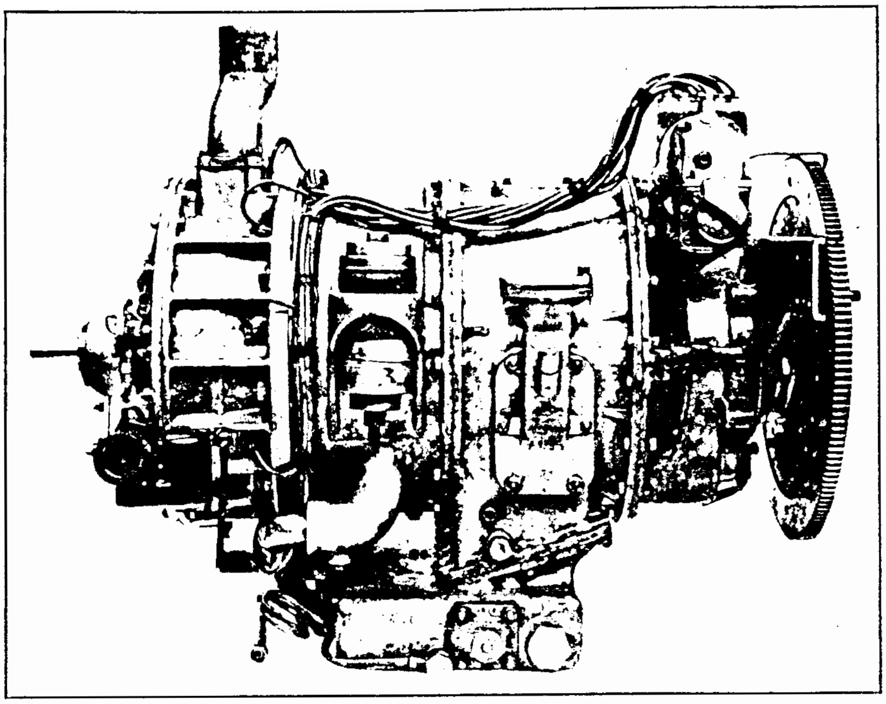


Fig. 17. The Bristol axial engine, Type R.R.3.

General proof

In the construction given and on the assumption that the radius of cylinder centres about the centre line of rotation and the diameter of operating centres in the swash plane are related :

$$K = R/2(1 + \cos\alpha)$$

then each such operating centre moves in a circle about the mating cylinder centre twice per revolution such that

$$r = R/2(1 - \cos\alpha).$$

The proof given, however, is partial and holds only for four selected maxima points. A more general proof follows.

In Fig. 8 let P Q S be the operative circle of the swash plate, first set perpendicular to the centre line of rotation o — radius R : and then at the operating angle α to the shaft as shown P T S.

Let P′Q′S′ be the circle of cylinder centres, in which $K = R/2(1 + \cos\alpha)$ as a special case.

Consider the shaft and swash plate at rest while the cylinders revolve anticlockwise through an angle θ from O Q.

Also consider that with each cylinder centre there moves over the swash plate a contact point C on the operative circle at an angular distance θ from the datum O T.

From the figure it follows that the point C on the swash plate, mating in the above manner with the cylinder centre B, is determined thus :—

$$AC = AD - CD$$

$$= R\cos\theta(1 - \cos\alpha)$$

$$AB = \text{a constant of design } K$$

$$BAC = \theta$$

$$\begin{aligned} CB^2 &= AC^2 + AB^2 - 2AC - AB\cos\theta \\ &= R^2\cos^2\theta(1 - \cos\alpha)^2 + K^2 \\ &\quad - 2K\,R\cos\theta(1 - \cos\alpha)\cos\theta \\ &= K^2 - R(1 - \cos\alpha)\,2K - R\,(1 - \cos\alpha)\cos^2\theta \end{aligned}$$

If $K = R/2(1 - \cos\alpha)$ it is clear that the second term of the right-hand expression becomes zero for all values of θ for

$$CB^2 = K^2 - 2K(2K - 2K)\cos^2\theta$$

hence the point C moves round the cylinder centre B in a circle of radius $R/2(1 - \cos\alpha)$.

If K be some other value then the resulting path will not be circular.

APPENDIX II

It was proven in Appendix I that the projection of the path of any point on the periphery of a wobble plate on a plane with normal C C′ in Fig. 7 is a circle of radius.

$$r = R/2(1 - \cos\alpha)$$

The curve whose co-ordinates are :—

$$x = s/2\cos nt.$$

$$y = r\sin 2\,nt.$$

where $s/2 = R\sin\alpha$

and $r = R/2(1 - \cos\alpha)$

satisfies this condition.

This curve is the resultant of the composition of 2 S.H.M.s, one of which has a frequency, double that of the other and which are mutually at right angles.

This may be tested, using the following data from the Nevatt " A " type engine.

$$R = 3{\cdot}833\text{in.}$$

$$K = 3{\cdot}6875$$

$$s/2 = 1{\cdot}458\text{in.}$$

The condition to give a circle is that

$x = 0$ gives a max. value of y.

i.e. $x_{y\,max.}$ (the value of x making y a maximum)

$$\begin{aligned} &= \sqrt{R^2 - K^2} \\ &= \sqrt{3{\cdot}833^2 - 3{\cdot}6875^2} \\ &= 1{\cdot}034\text{in.} \end{aligned}$$

y is a max. when $dy/d\,(nt) = 0$

$$dy/d\,(nt.) = 2nr\cos 2\,nt.$$

$$= 0 \text{ to give } y_{max.}$$

$$\therefore\ nt_{y\,max.} = \pi/4$$

$$\therefore\ x_{y\,max.} = 1{\cdot}458\cos\pi/4.$$

$$= 1{\cdot}034, \text{ giving an exactly identical result}$$

Values of x and y for one-quarter of the complete curve are appended below.

nt deg.	cos nt.	x $s/2$ cos nt.	sin 2 nt.	y r sin 2 nt.
0	1·000	1·458	0	0
10	0·9848	1·416	0·3420	0·0492
20	0·9397	1·380	0·6428	0·0938
30	0·8660	1·271	0·8660	0·1263
40	0·7660	1·121	0·9848	0·1437
45	0·7071	1·031	1·0000	0·1460
50	0·6428	0·941	0·9848	0·1437
60	0·5000	0·731	0·8660	0·1263
70	0·3420	0·502	0·6428	0·0938
80	0·1736	0·255	0·3420	0·0492
90	0	0	0	0

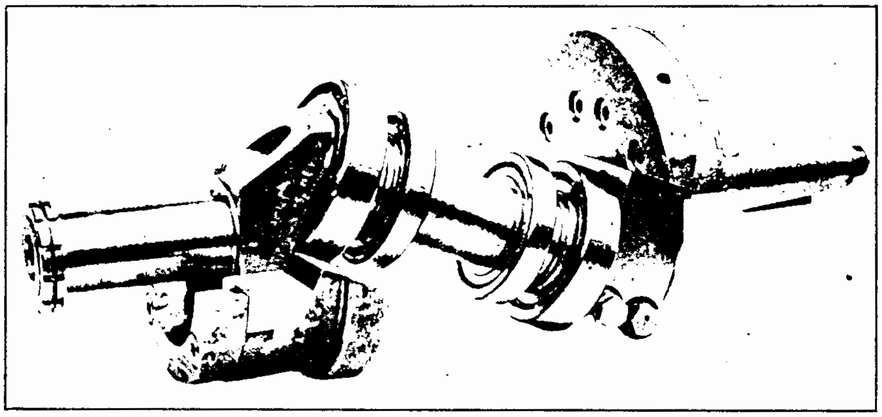


Fig. 18. Crank assembly, Type R.R.4.

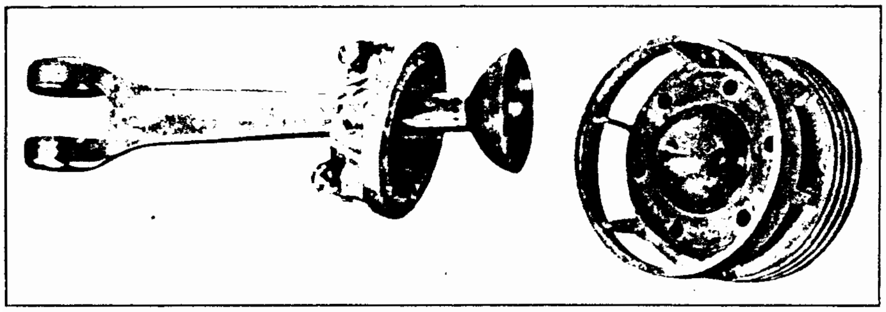


Fig. 19. Piston ball joint for Bristol engine, Type R.R.2.

APPENDIX III

Calculation of engine torque

In order to simplify the calculation and render it more applicable it is necessary to consider that the ball joint has a straight line motion along the cylinder centre. This has very little effect on the accuracy of the result, since the chord of the lemniscate for a wobble angle of 22½ deg. is less than one-tenth of its length. The arm of the torque, S, is greatest at half-stroke and zero at T.D.C. and B.D.C. The position of the stabiliser roller for the cylinder in question is at its highest point in the stabiliser casing at half-stroke. The side thrust at this point is therefore being taken by the stabiliser 90 deg. in advance of this particular one. In practice, however, the total torque reaction from all cylinders is delivered to the crankcase by those cylinders whose stabilisers are on the thrust side of their casings.

The piston thrust at any position of the crank can be resolved into two forces T & W, T being the side thrust on the stabiliser and W the thrust on the ball-connector of the wobble plate. (See Fig. 11.) W is in the vertical plane through the crankpin centre line and coplanar with P and T.

At half-stroke

$$W = P \sec \alpha$$
$$T = P \tan \alpha$$

where α is the angle of wobble.

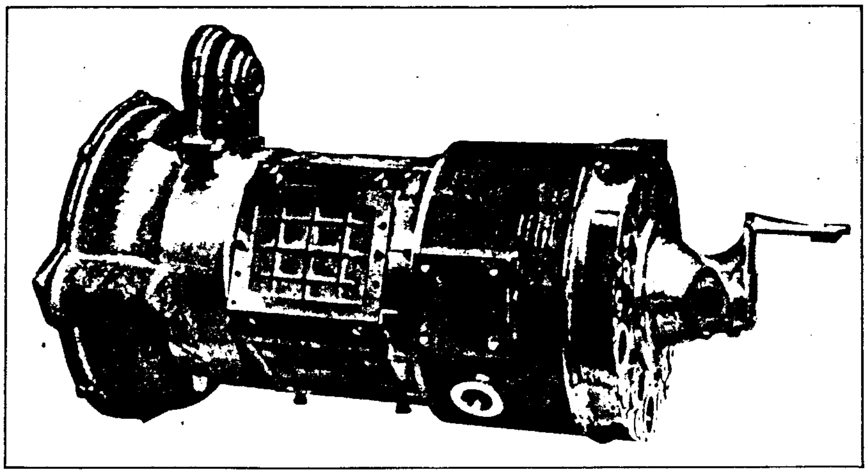


Fig. 20. Nevatt axial engine, Type A.

For intermediate positions

$$W = P \sec \alpha_1$$
$$T = P \tan \alpha_1$$

where $\alpha_1 = \alpha \sin \theta$
and θ = crank angle after T.D.C.

Since the mean radius at which T is exerted is K, the radius of cylinder centres, the cylinder torque at any given crank angle position is given by

$$T_c = K P \tan (\alpha \sin \theta)$$

From this equation the torque delivered by the cylinder can be evaluated for the whole cycle and when the torque curves of all cylinders are combined the actual curve of total torque is obtained.

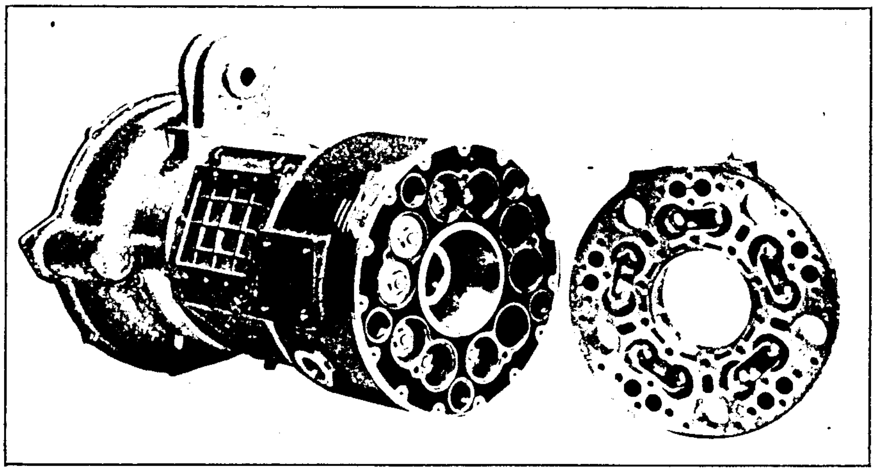


Fig. 21. Nevatt engine with cylinder head removed.

APPENDIX IV

Balancing the wobble plate engine

The motion of the wobble plate produces a rotating couple which is in perfect phase with the rotating couple produced by the inertia of the reciprocating weights.

By arranging two suitable balance weights, as in Fig. 14, an equal and opposite rotating couple is obtained. Apart from the slight lateral movement occasioned by the figure eight path of the wobble plate, it is possible to achieve perfect balance in this type of engine.

Inertia force due to wobble plate

Fig. 12 shows section through a simple form of wobble plate. Consider small increment of area δA at radius R. This increment may be considered as having a simple harmonic motion of maximum displacement R sin ϕ where ϕ is the angle of swash. This maximum occurs when the crank angle, θ, measured from T.D.C. = 0 deg. or 180 deg.

Max. displacement, $r = R \sin \phi$

$\therefore$ Inertia force $= W\omega^2 r \cos \theta$ poundals.

$$= \frac{W}{g}\left(\frac{\pi N}{30}\right)^2 R \sin \phi \cos \theta \text{ lb.}$$

where R is in feet.

For all practical purposes the inertia force acting on one cylinder may be taken as that due

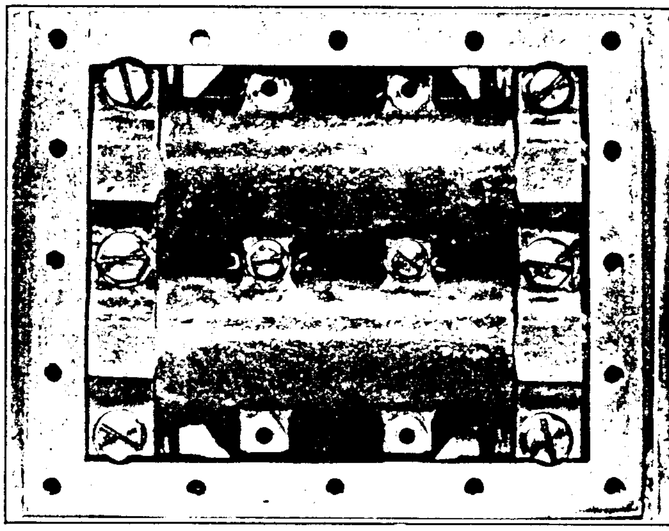


Fig. 22. View through crankcase inspection aperture.

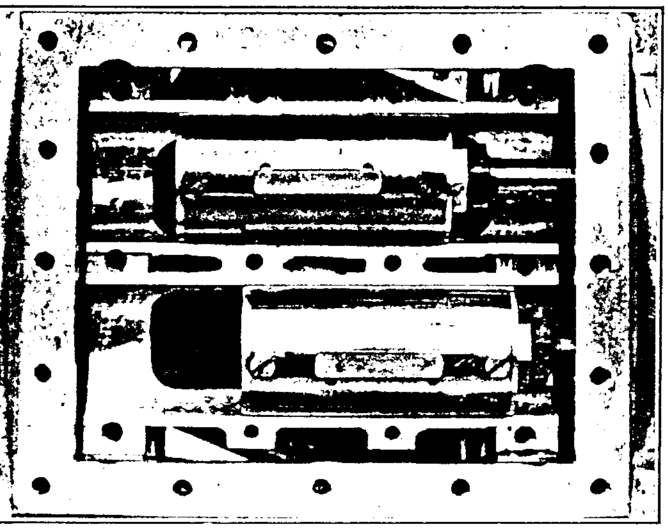


Fig. 23. View with top half of stabiliser guide removed.

to a segment of the wobble plate of angle $360/n$ where n = the number of cylinders.

The weight of each annular strip of metal may be considered as concentrated at one point if the number of cylinders is large. Thus the volume of the strip of area δA in cross-section

$$= \frac{2\pi R}{n} \times \delta A$$

The volume at different radii can be drawn by graphical methods. In Fig. 13, A^2 represents the weight of the segment whose cross-section is A_1. Each ordinate of A_2 is equal to $\frac{2\pi R\delta}{n}$ times the ordinate of A_1 where δ is the density, i.e.,

$$y_2 = \frac{2\pi R\delta}{n} y_1$$

Similarly A_3 may be drawn by multiplying each ordinate of A_2 by

$\frac{R}{g}\left(\frac{\pi N}{30}\right)^2 \sin\phi$ where R is in feet.

The weights of any extra pieces, such as the ball connector and its fixing bolts, should be added to A_3.

The total moment of $1/n$th part of the wobble plate can be found in this way.

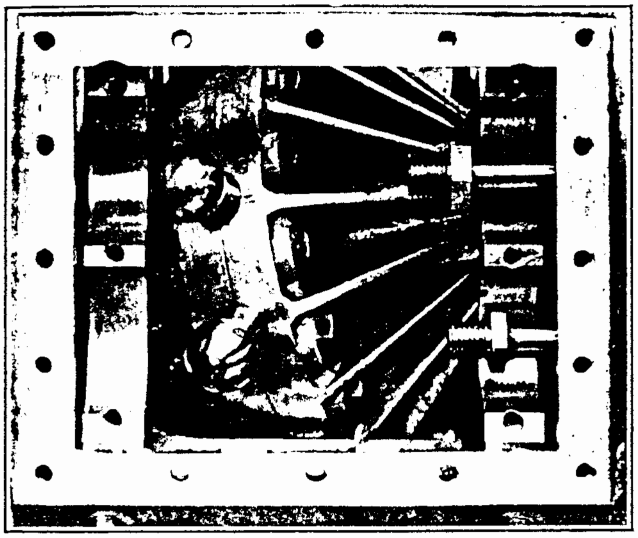


Fig. 24. View with stabiliser guide completely removed.

The rotating couple due to the wobble plate inertia is composed of this moment, M, due to the maximum inertia of $1/n$th part of the wobble plate, together with less inertia forces due to the other segments of the wobble plate which are not at their maximum displacement position. The total moment on the shaft from the wobble plate is given by

$$M_T = M\,(1 + 2\cos^2\theta + 2\cos^2\theta_2 + \ldots + 2\cos^2\theta_n)$$

where $\theta = \frac{360}{n}$; $\theta_2 = 2\theta$; $\theta_3 = 3\theta$

and θ_n is less than $\pi/4$

It is seen from Fig. 14 that the total moment, $M_T = F_I R_G$

To give perfect balance

$$F_I R_G = F_w R_w$$

$$F_w = \frac{w}{g}\left(\frac{\pi N}{30}\right)^2 r$$

$$\therefore\ M^T = \frac{W r R w}{g}\left(\frac{\pi N}{30}\right)^2$$

$$W R_w r = F_I R_G g \left(\frac{30}{\pi N}\right)^2$$

This gives the required balance weight when the values R_w and r are chosen to fit the particular design.

Cover sheet for a 21 page patent

United States Patent [19]

Searle

[11] **4,138,930**

[45] **Feb. 13, 1979**

[54] **PISTON AND CYLINDER MACHINES**

[76] Inventor: **Russell J. Searle,** 20 Park St., Windsor, England

[21] Appl. No.: **574,713**

[22] Filed: **May 5, 1973**

[30] **Foreign Application Priority Data**

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May 6, 1974	[GB]	United Kingdom	19784/74
Nov. 6, 1974	[GB]	United Kingdom	47989/74
Nov. 21, 1974	[GB]	United Kingdom	50496/74
Nov. 26, 1974	[GB]	United Kingdom	51163/74
Dec. 5, 1974	[GB]	United Kingdom	52629/74

[51] **Int. Cl.**[2] **F01B 3/00; F02B 57/00**

[52] **U.S. Cl.** .... **92/70;** 91/501; 92/57; 123/43 A

[58] **Field of Search** .... 91/501, 499; 92/57, 92/70, 71; 123/43 A, 58 R, 58 B, 58 BA, 18 R, 193 CP; 417/269

[56] **References Cited**

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Primary Examiner—Carlton B. Croyle

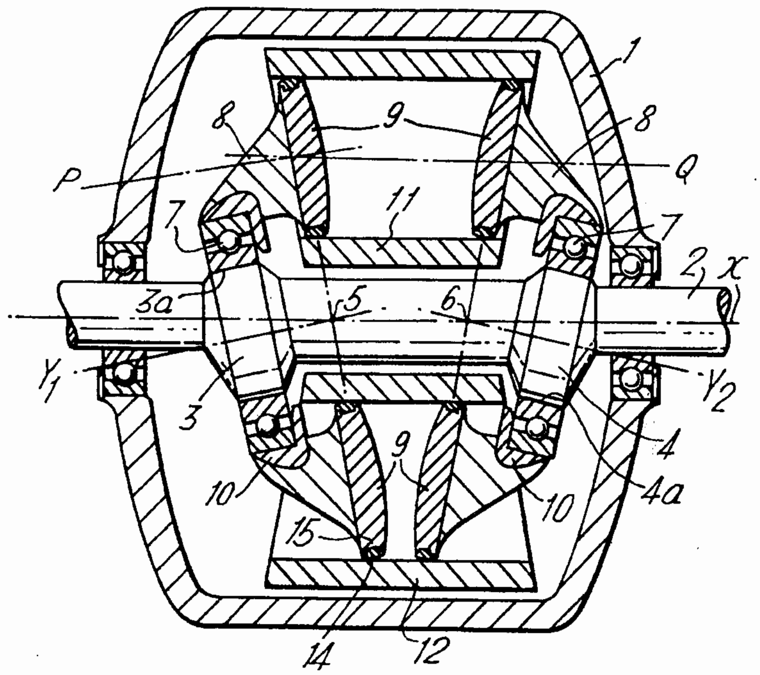
Assistant Examiner—Michael Koczo, Jr.

Attorney, Agent, or Firm—Haseltine, Lake & Waters

[57] **ABSTRACT**

A piston and cylinder machine of the kind in which there is a wobble mechanism for reciprocating the piston in the cylinder or an arrangement in which a piston assembly and a cylinder assembly rotate about respective inclined axes so that such rotation is accompanied by relative reciprocation of the piston and cylinder, is provided with a piston having a part which has a surface exposed inside the cylinder to define at least a part of one end of a chamber therein and which, to provide sealing of the chamber, has a peripheral surface forming a sphere. Improved torque transmitting or reacting means for maintaining the correct relative orientation of the piston and cylinder are provided.

19 Claims, 16 Drawing Figures



See Popular Science Oct. 1981 p. 29-30

United States Patent [19]

Searle

[11] **4,363,294**

[45] **Dec. 14, 1982**

[54] **PISTON AND CYLINDER MACHINES**

[76] Inventor: **Russell J. Searle**, 1, Thames Corner, Thames Street, Sunbury-on-Thames, Middlesex, England

[21] Appl. No.: **41,902**

[22] Filed: **May 22, 1979**

[30] **Foreign Application Priority Data**

May 25, 1978 [GB] United Kingdom 22849/78
Dec. 14, 1978 [GB] United Kingdom 48553/78

[51] **Int. Cl.**[3] **F02B 57/00**; F02B 75/26

[52] **U.S. Cl.** .... **123/43 A**; 123/58 BA; 123/193 P; 92/70; 92/57; 277/138

[58] **Field of Search** .... 123/43 R, 43 A, 43 C, 123/58 R, 58 A, 58 AA, 668, 669, 18 R, 193 P; 277/138; 92/70, 57

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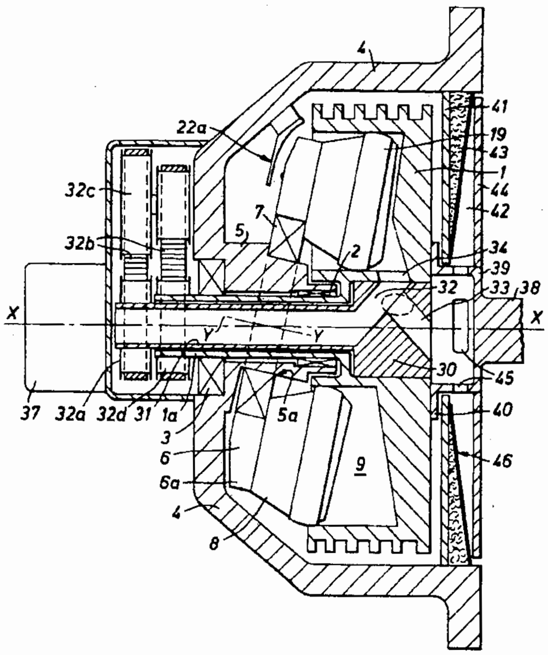
4,138,930	2/1979	Searle	123/43 A

Primary Examiner—Ronald B. Cox
Attorney, Agent, or Firm—Murray and Whisenhunt

[57] **ABSTRACT**

A piston and cylinder machine, e.g. an engine or pump, in which the cylinder rotates about a first axis from which the cylinder is spaced, and the piston rotates about a second axis inclined to the first with the result that during the rotation, the piston reciprocates relative to the cylinder. Gas flow is controlled by a rotary valve which communicates with the cylinder and which rotates about an axis parallel to or coincident with the first axis. The piston is maintained in position relative to the cylinder by restraining means in the form, for example, of flat abutment surfaces provided between the piston and a spherically-surfaced piston ring thereon so as to limit lateral movement of the piston ring relative to the piston.

6 Claims, 9 Drawing Figures



see Popular Science, Oct. 8, p. 29-30

FIG. 1.

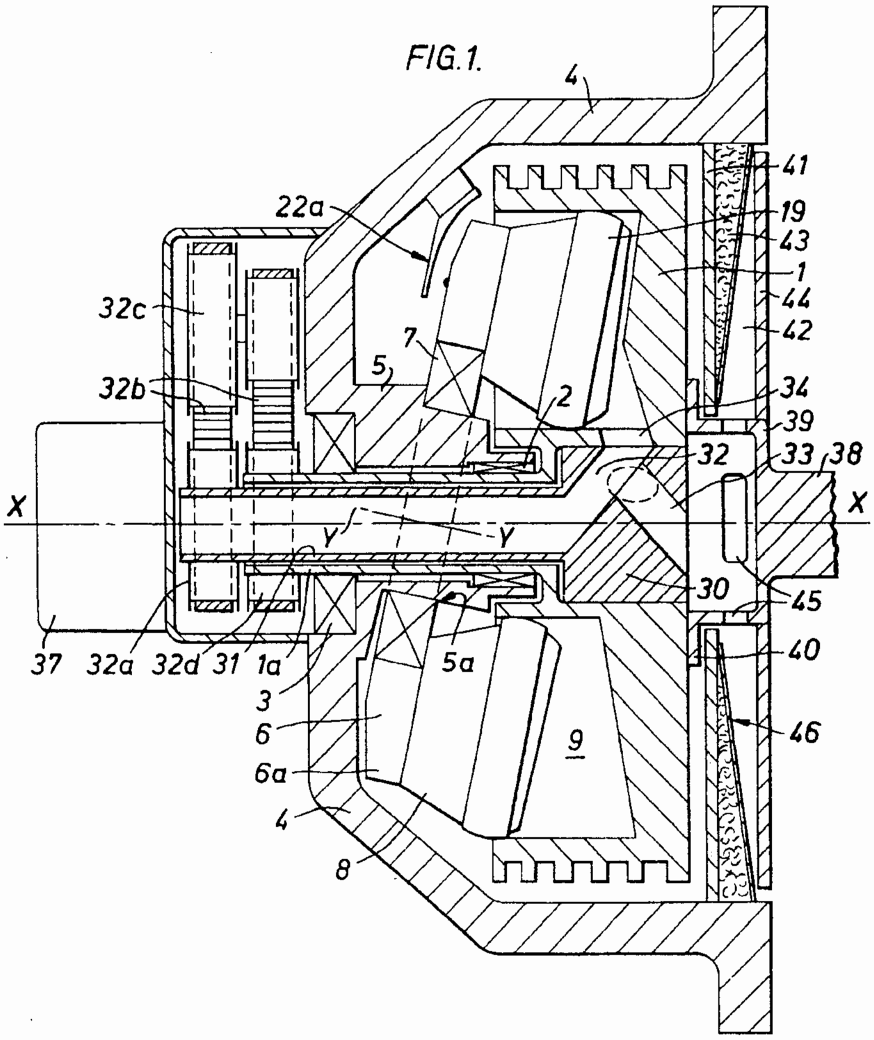

4
41
22a
19
43
1
32c
44
42
7
5
2
32b
34
39
32
33
38
X
Y
Y
X
30
45
37
32a
32d
31
1a
5a
40
3
6
46
9
6a
4
8

FIG. 2.

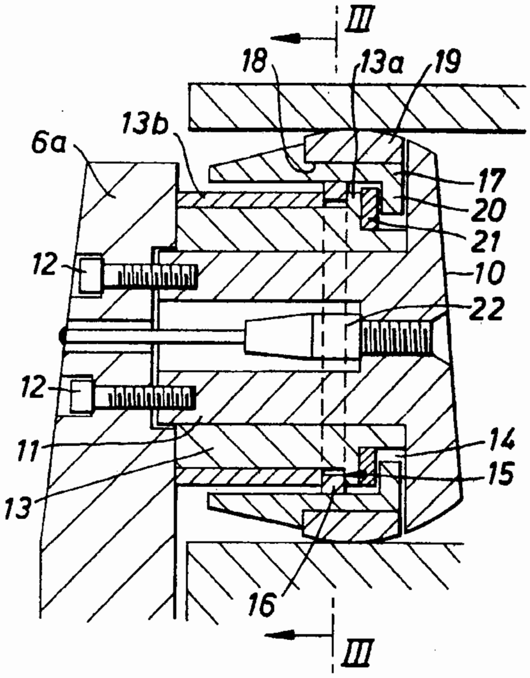


FIG. 4.

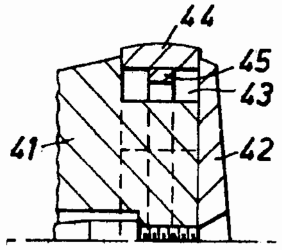


FIG. 3.

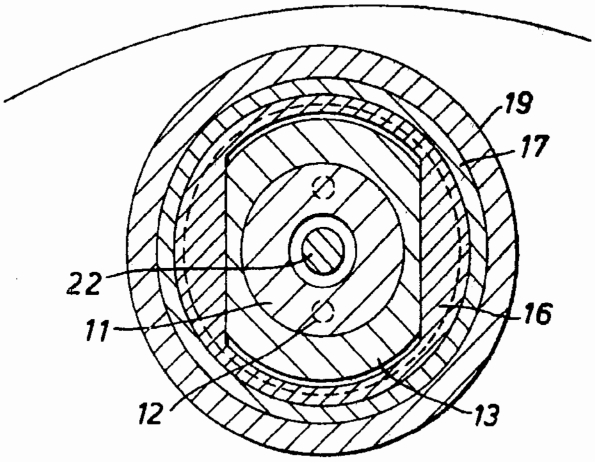


FIG.2.

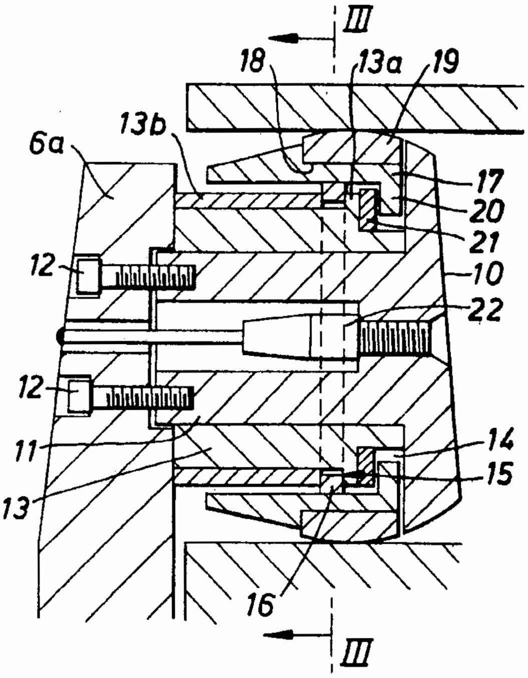


FIG.4.

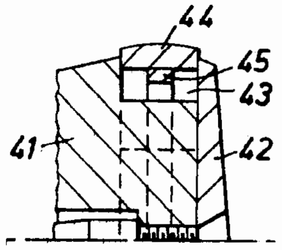


FIG.3.

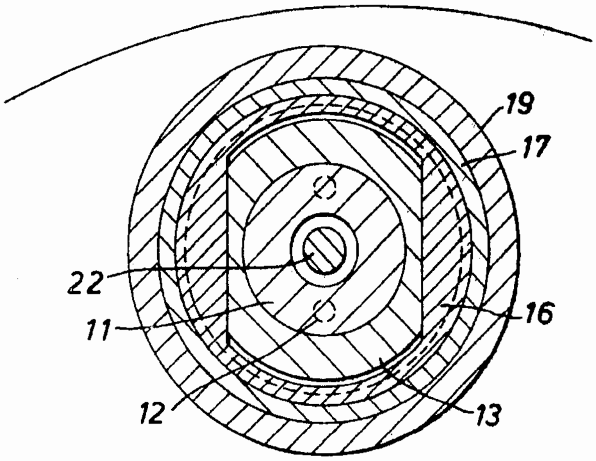


FIG.5a.

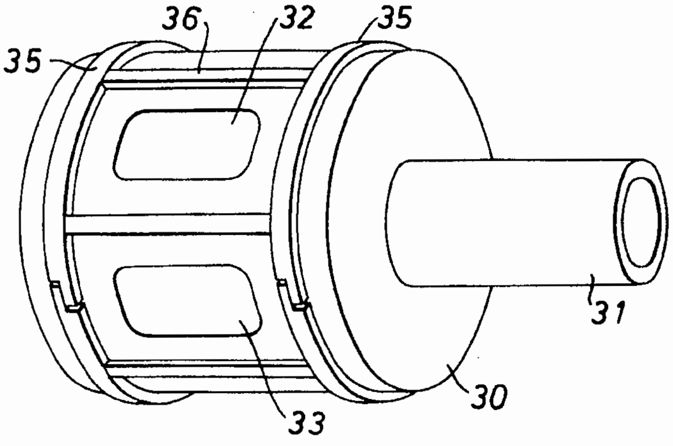


FIG.5b.

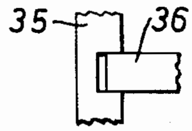


FIG.5c.

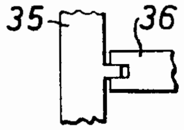


FIG.5d.

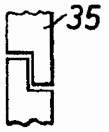
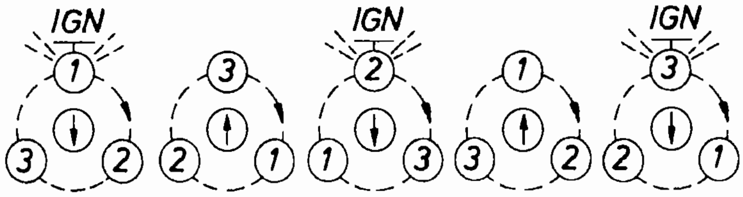


FIG.6.



ROTATION					
CYLINDERS	0	120	240	360	480
VALVE	0	180	360	540	720

PISTON AND CYLINDER MACHINES

This invention relates to a piston and cylinder machine, e.g. an engine or pump.

According to the invention, there is provided a machine comprising:

a cylinder having a straight sided bore spaced from a first axis,

a piston assembly including a piston which is disposed within said cylinder bore to form a chamber therein, the piston assembly further including a connecting portion connected to the piston and restraining means for resisting any tendency for the piston to move in relation to the cylinder about the first axis,

an inclined member mounted for relative rotation between the inclined member and the cylinder to occur about the first axis, the inclined member being inclined to the first axis along a second axis and the said connecting member being engaged with the inclined member such that said relative rotation of the cylinder and the inclined member is accompanied by reciprocation of the piston within the cylinder, and

a rotary valve which communicates with said chamber and which rotates about an axis parallel to or coincident with said first axis.

For a better understanding of the invention, and to show how the same may be carried into effect, reference will now be made, by way of example, to the accompanying drawings, wherein:

FIG. 1 is a diagrammatic partly sectional elevation of an engine,

FIGS. 2 and 3 are a sectioned side view and a sectioned plan view respectively of a piston used in the engine of FIG. 1,

FIG. 4 is a sectioned side view of a part of a modified piston,

FIG. 5*a* is a perspective view of a valve used in the engine of FIG. 1,

FIGS. 5*b* to 5*d* are explanatory diagrams relating to FIG. 5*a*, and

FIG. 6 is a series of diagrams showing the operating sequence of a valve used in the FIG. 1 engine.

The engine of FIG. 1 comprises a cylinder assembly including a cylinder block **1** formed with an axially extending hollow shaft portion **1***a*. The shaft portion **1***a* is mounted in bearings **2** and **3** so that the whole cylinder assembly can rotate about an axis X. The cylinder block **1** is enclosed in a stationary casing **4** having a hollow projecting portion **5** which extends inwardly from one end of the casing and which forms a housing for the bearings **2** and **3**. Part of the portion **5** defines a cylindrical surface **5***a* of which the axis y is inclined to the axis x. A piston assembly **6** is supported on this surface by way of a bearing **7** such that the piston assembly can rotate about axis y. The piston assembly comprises a connecting member **6***a* which is mounted at its centre on the bearing **7** and which has three radially extending portions to each of which is attached a piston **8**, each piston being disposed within a respective cylinder bore **9** in the cylinder block **1**. Rotation of the cylinder assembly about the axis X is accompanied by rotation of the piston assembly about the axis Y and, since axes X and Y are inclined to one another, such rotation is further accompanied by reciprocation of each piston relative to its respective cylinder bore.

As mentioned there are three pistons and correspondingly, the cylinder block has three bores, these being arranged equidistant about axis of rotation X. In FIG. **1** to better illustrate the construction, only two pistons and two bores are shown spaced at 180° whereas in fact there are three and they are spaced at 120° intervals.

As shown in FIGS. **2** and **3**, each piston **8** comprises a part defining a sloping piston crown **10** and a spigot **11** which extends back from the crown **10** and is attached by screws **12** to the connecting member **6***a*. Disposed around the spigot **11** between the connecting member **6***a* and the crown **10** is a cylindrical member **13** of which the end next to the crown **10** is reduced in diameter so as to form shoulder which, with the rear of the crown **10**, defines a circular recess **14**. A short way back from this end of the member **13**, there is an annular flange **13***a* and round the member **13** there is a further cylindrical member **13***b* which has the same external diameter as the flange **13***a*, and which extends from the other end of the member **13** up to a short distance from the flange **13***a* such that there is defined a groove or channel **15** between the flange **13***a* and member **13***b*. In this groove **15** there is disposed a ring-shaped member **16** with clearance between the inner surface of the ring-shaped member **16** and the floor of the groove **15** so that the member **16** can move as a whole laterally with respect to the member **13**. In one and only one of the pistons **8**, the groove **15** and member **16** are as shown in FIG. **3**, that is with two opposite "flats" and only sufficient clearance between the flat portions of the inner surface of the member **16** and the flat portions of the floor of the groove **15** to allow sliding movement. Thus, for this piston **8**, the movement of the member **16** with respect to the member **13** is restricted to the directions towards and away from the axis Y in FIG. **1**. Fitted closely around the member **16** such that there is clearance between it and the member **13**, is a collar **17** which is formed with a recess **18** for receiving a continuous piston ring **19** (i.e. a ring which is continuous all around its circumference instead of being split as in conventional piston rings) and with an inwardly directed flange **20** which extends into the recess **14**. Between the flange **20** and that wall of the recess **14** which is formed by the shoulder of the member **13** there is disposed a thrust bearing spacer annulus **21** to take the thrust exerted by the piston ring **19** via the collar **17**. The piston ring **19** has an external periphery in the shape of an equatorial region of a sphere of diameter equal to that of the cylinder bore. An ignition plug **22** is screwed into a threaded bore formed in the piston crown **10** such that the plug can ignite fuel/air mixture within the cylinder. It will be appreciated from a consideration of FIG. **1** that, as each piston **8** reciprocates with respect to its cylinder, it moves along an arcuate path such that its distance from axis X changes during the reciprocation. This change in distance is taken up by the lateral movability as a whole of the ring-shaped member **16** and hence also of the collar **17** and piston ring **19**, each as a whole, with respect to the member **13**. The limitation of this movement, for one of the pistons, to the direction towards and away from the axis X ensures that the piston assembly is restrained from rotation in relation to the cylinder block and the pistons remain substantially central within the cylinder bores.

Each piston crown **10** and/or the end surface of each cylinder could be coated with ceramic material to reduce heat loss from the combustion chamber.

The construction of the piston may be modified in various ways. For example, instead of having the relatively complex internal structure shown in FIGS. **2** and

3, it could comprise a simple main body part **41** and a removable crown **42** as shown in FIG. **4**, the body part **41** being shaped such that between it and the crown **42** there is formed a groove **43** for the piston ring **44**. The floor of the groove is circular but with two opposite "flats", i.e. the same shape as the groove **15** in FIG. **3**. Then, for one of the pistons, there is provided an intermediate ring-shaped member **44** which also has two opposite "flats" on its internal surface which mate with the flats of the floor of the groove **43** and limit the relative movement of the piston ring **44** of this piston to the directions towards and away from the axis X in FIG. **1**. The intermediate ring **44** is not provided in the other pistons so that, here, the piston ring can move in all directions laterally with respect to the piston.

Instead of being continuous, the piston ring could be split at a point on its circumference in the manner of a conventional piston ring. Then it is preferred that the split should follow a diagonal, cranked or other tortuous path across the width of the ring so as to reduce the possibility of gas leakage through the split. The piston ring **19** could be made of synthetic plastics material or metal, or metal which is coated with synthetic material or plated with another metal. The main criteria is that the material forming the surface which contacts the cylinder should be compatible with the cylinder material or the material of the cylinder liner if these are provided. If the chosen ring material is such that allowing for relative coefficients of expansion of this material and the cylinder material and the different temperature rises which they may undergo during operation of the engine, the ring material will expand more than the cylinder, it may be necessary to give the ring a composite construction. For example, each piston ring **19** may consist of two concentric rings one within the other and bonded or shrunk together the outermost ring being profiled to form the equatorial zone of a sphere of a diameter equal to the diameter of the cylinder. Then, to maintain a constant sealing clearance between the profiled ring and the cylinder, the coefficients of expansion of the two rings and the cylinder are chosen so that the following condition applies:

$$D_1X_1t_1 = D_2x_2t_2 + 2wx_3t_3$$

where t1, t2 and t3 are the respective temperature rises of the cylinder and the inner and outer rings during operation of the engine, D_1 and D_2 are the diameters of the cylinder bore and the inner ring respectively and,

x_1 = coefficient of expansion of cylinder
x_2 = coefficient of expansion of inner ring
x_3 = coefficient of expansion of outer ring
w = radial thickness of inner ring.

The function of the inner ring could be taken over by the collar **17**, i.e. the ring itself being made in one piece and of material having a coefficient of expansion x_3 and the material of the collar **17** having a coefficient x_2. A combined rotary inlet and exhaust valve **30** is mounted in a valve housing formed in the cylinder block **1**. The valve has a tubular extension **31** which passes through and is rotatable in the hollow shaft **1***a* of the cylinder block..

Referring to FIG. **5***a* as well as FIG. **1**, the valve itself comprises a cylindrical member with ports **32** and **33** formed therein. Port **32** leads from the interior of the tubular extension **31** to one position on the cylindrical surface of the valve and the port **33** leads from another position on this surface to the end of the valve furthest from the extension **31**. A port **34** in the wall of each cylinder leads to the valve so that each cylinder port **34** communicates periodically with each of the ports **32** and **33**. Fuel/air mixture is made available to the port **32** via the interior of the extension **31** and exhaust gases are emitted via port **33**. That end of the extension **31** which protrudes from the end of the cylinder block shaft portion **1***a* is fitted with a toothed pulley wheel **32***a* which is coupled via two toothed belts **32***b* and two intermediate pulleys **32***c* to a pulley **32***d* which is fitted to the shaft portion **1***a*. The pulleys **32***c* are of such relative size that the valve **30** is rotated at one and a half times the speed of the cylinder block. To maintain the belts in position on the pulleys **32***a*, **32***c* and **32***d* and to prevent the belts from jumping teeth on the pulleys and thereby changing the relative positions of the different pulleys, spring mounted idler pulleys (not shown) are arranged to press on the outer surfaces of the belts at appropriate positions near the other pulleys.

Because of the relative speeds of the cylinder block and valve, gas is induced, compressed and ignited, expanded and exhausted in each cylinder once per two revolutions of the system, the positions on the surface of the valve where the ports **32** and **33** open being such as to give correct timing of these functions. FIG. **6** shows the relative positions of the rotary valve as each cylinder moves into the top dead centre position. It requires two revolutions after the first position shown i.e. ignition in cylinder **1**, for the valve to be again in the correct position for ignition in cylinder **1**.

At each end of the valve **30** there is a groove containing a spring ring **35**. Extending between the two rings **35**, one on that side of each of the ports **32** and **33** which is furthest from the other port and one between the two ports, are lengthwise grooves each containing a sealing strip **36**, each end of each strip being engaged with the adjacent ring **35** so as to allow differential expansion of the rings and strips while enabling the strips to be maintained in position and to be sprung outwardly by the rings **35** against the surface of the valve housing formed in the cylinder block. For example, the strips may be engaged with the rings by the means shown in FIGS. **5***b* and **5***c*. Each ring is split, the ends at the split being stepped as shown in FIG. **5***d*. The rings and strips provide "piston ring" type sealing between the valve and the valve housing. As an alternative to what is shown, the rings and strips could be mounted in grooves in the internal surface of the valve housing and arranged to spring inwardly onto the valve, in which case only two strips are necessary.

The carburettor **37** is stationary and communicates with the inlet passage through the tubular extension **31** of the rotary valve via a rotating seal, inlet gases passing through the inlet passage in the valve and into the cylinder port **34**. The valve remains open during the induction stroke and is closed during the compression and ignition stroke. The exhaust passage **33** in the valve communicates with the cylinder port **34** during the exhaust stroke.

The engine drive shaft **38** has a hollow, enlarged end **39** which is fixed by means of a flange **40** to the cylinder block around the valve housing therein. Fixed within the casing **4**, around the enlarged end **39**, is a partition plate **41** which bounds one side of a chamber **42** containing sound deadening material **43** such as glass or wire wool. The other side of the chamber is bounded by a plate **44** fixed to shaft **38**. The diameter of the plate **44** is such that there is left a gap all round the periphery of the chamber. Exhaust gases pass from the port **33** into

the hollow end **39** of shaft **38** and then via radial ports **45** in this end to the chamber **42** where they pass over the sound deadening material **43** and thence to the exterior via the open periphery of the chamber. The sound deadening material **43** is held in place by a dished member **46** made of perforated metal for example.

Various alternative exhaust systems could be used instead of the particular one shown. For example, a stationary exhaust manifold may communicate with the port **33** by way of a sliding or labyrinth seal, there then being provided a stationary silencer which may be remote from the engine, or the plate **41** in the illustrated arrangement could be replaced by a shallow cylindrical dish fixed to the shaft portion **39** instead of to the casing **4** so that the whole of the silencer system rotates with the shaft.

The ignition plugs **22** communicate cyclically with an electrical contact **22a** fixed to the casing. The ignition make and break (not shown) is operated on a single lobe cam from an extension to the shaft carrying the idler timing pulleys or gears **32** which if arranged to rotate at valve speed i.e. $1\frac{1}{2}$ time engine speed will produce a spark at each revolution of the valve and ignite consecutively cylinders **1**, **2** and **3** as shown in FIG. **5**.

The rotary valve and its housing are made of material of a similar coefficient of expansion, for example aluminium alloy, one valve sealing surface being coated with a high temperature synthetic polymer, or a metal mixed with synthetic polymer, graphite, molybdenum disulphide or other lubricative material, the other hardened by anodising or having a coating of metal, ceramic or synthetic polymer compatible with the mating surface. The coated surfaces may be sprayed, for example, by means of a plasma arc gun, with particles of the coating material or alternatively the coating may be plated or deposited chemically or mechanically onto the valve sealing surface. The coating is sufficiently thin to ensure that over the temperature range to which the valve is subjected the sealing clearance between the valve and its housing remains substantially constant.

Air cooling ducts may be formed in the cylinder block comprising for example three parallel holes intermediate to the cylinder bores and adjacent to the rotary valve which communicate with a series of radial holes passing to the outer periphery of the block, thus air is centrifuged out through the radial holes and drawn in along the parallel holes.

The engine could have a number of cylinders other than three the relative speed of rotation of the rotary valve and cylinder block then being appropriately set to give two or four stroke operation of the engine as desired. For a three cylinder engine only one of the piston rings is restricted in its lateral movement with respect to its associated piston, the other two rings being able to move in all directions laterally so as to take up the relative sideways changes in position which occur between the associated pistons and cylinders. In an engine having other than three cylinders it may be possible for say two of the piston rings to be restricted in the direction in which they move this depending upon the geometry of the relative movement between piston and cylinder assemblies of the engine concerned.

Furthermore, instead of or in addition to this illustrated means for preventing circumferential movement of the pistons relative to the cylinders about the axis of rotation of the engine by restricting the piston ring movement, i.e. the flat parallel portions in the piston and the correspondingly shaped intermediate member as shown in FIG. **3**, the piston rotor and cylinder block could be coupled together by way of a linear bearing arrangement as disclosed in UK patent specification No. 1,511,232 for example, the arrangements comprising the members 103, 104 and such illustrated in FIG. **10** of the drawings of that specification.

Instead of being as shown, the cylinder block of the engine could rotate about axis Y and the pistons about axis X.

Furthermore, instead of the piston(s) and cylinder(s) rotating it could be the member **5** in FIG. **1** which rotates, the member **5** instead of being formed as part of the engine casing being coupled to an engine output shaft for example to rotate with it. It will be appreciated that the basis operation of the engine is still the same since it is the relative rotation of the member **5** and the piston/cylinder assembly which produces or which is produced by the reciprocation of the pistons in the cylinders. Machines wherein the support member rotates and the piston/cylinder assembly does not are disclosed in specification No. 1,511,232 and the present invention includes the modification of any of those machines to include a rotary valve of the kind disclosed herein. It may of course be necessary to modify the arrangement for driving the rotary valve and/or to modify its speed of rotation.

The construction illustrated could be adapted to form a pump. For example the ignition plugs carburettor and exhaust silencer system could be discarded and a drive motor provided to rotate the shaft **38**.

What I claim is:

1. A machine comprising:

(a) a cylinder having a straight sided cylinder bore spaced from a first axis;

(b) a piston assembly including a piston shaped to define a peripheral recess having a floor with two opposite flat portions, said piston being disposed within said cylinder bore to form a chamber therein, said piston assembly further including a connecting member connected to said piston and restraining means for resisting any tendency for the piston to move in relation to said cylinder about said first axis;

(c) an inclined member mounted such that relative rotation between said inclined member and said cylinder occurs about said first axis, said inclined member being inclined to said first axis along a second axis, said connecting member being coupled to said inclined member such that said relative rotation of said cylinder and said inclined member is accompanied by reciprocation of said piston within said cylinder;

(d) valve means for fluid moving through said machine in the use thereof;

said piston assembly also including a piston ring disposed around said piston, said piston ring being moveable as a whole laterally with respect to said piston, and said restraining means comprising a ring-shaped member having two flat internal surface portions which match said two opposite flat portions of said floor of said recess, said ring-shaped member being engaged in said recess in such a way that lateral movement of said piston ring relative to said piston is restricted to a to-and-fro direction which is radial, or which has a major component which is radial, with respect to said second access.

2. A machine comprising:

(a) a cylinder having a straight sided cylinder bore spaced from a first axis;

(b) a piston assembly incliding a piston disposed within said cylinder bore to form a chamber therein, said piston assembly further including a connecting member connected to said piston and restraining means for resisting any tendency of said piston to move in relation to said cylinder about said first axis.

(c) an inclined member mounted such that relative rotation between said inclined member and said cylinder occurs about said first axis, said inclined member being inclined to said first axis along a second axis, said connecting member being coupled to said inclined member such that said relative rotation of said cylinder and said inclined member is accompanied by reciprocation of said piston within said cylinder;

(d) valve means for fluid moving through said machine in the use thereof;

said piston assembly also including a piston ring disposed around said piston; said piston ring being moveable as a whole laterally with respect to said piston, said piston ring having a composite structure consisting of two concentric ring-shaped members which adhere to one another, and which are made from materials of differing coefficients of expansion so as to obtain an effective overall coefficient of expansion which is such that the piston ring and the cooperating cylinder expand by substantially the same amount during operation of the machine, and said restraining means comprising means for restricting the lateral movement of said piston ring relative to said piston to a to-and-fro direction which is radial, or which has a major component which is radial, with respect to said second axis.

3. A machine according to claim **1** or **2**, wherein said piston ring has a periphery in the shape of an equatorial region of a sphere whose radius is equal to that of the cooperating cylinder.

4. A machine according to claim **1** or **2**, wherein at least a surface of said piston ring is made of synthetic plastics material.

5. A machine according to claim **1** or **2**, wherein said valve means comprises a rotary valve which communicates with said chamber, said valve and said cylinder being disposed for relative rotation, in use, about an axis parallel to or coincident with said first axis.

6. A machine according to claim **5**, wherein there are three cylinders spaced around said first axis, and wherein said piston assembly comprises three pistons disposed within respective ones of said three cylinders, said machine being constructed as a four stroke internal combustion engine and said rotary valve being arranged to rotate about said first axis at one-and-a-half times the speed of the machine and to communicate sequentially with said three cylinders.

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